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AN EVALUATION OF HYDRODYNAMIC
FLUID CLUTCHES FOR SPACE APPLICATIONS

A THESIS

Presented to

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by

William Arthur Jones

In Partial Fulfillment

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SUMMARY

The possibility of the use of hydrodynamic fluid clutches in space applications is investigated in this work. The characteristics, capabilities, and limitations of these clutches are investigated theoretically and experimentally.

A general survey of fluid clutches was conducted and several characteristics obtained in the survey are discussed. Circuit modifications for the traction type, basic arrangements of the adjustable speed type, and selection tables for both the traction and adjustable speed are presented. Horsepower ratings, speed, approximate dimensions, and weight are given in the traction clutch table, while horsepower, speed, approximate dimensions, slip, and control force are presented in the adjustable speed clutch table. Torque-to-weight and torque-to-volume ratios for the traction clutch are also presented.

A small horsepower fluid clutch was experimentally tested to determine the effects on the clutch torque capacity of the following parameters: viscosity, fluid temperature, quantity of fluid in the clutch, input speed, slip, and clearance between the clutch impeller and turbine. Two petroleum-based oils were used in the tests. One oil had a viscosity of about 200 Saybolt Universal seconds, while the other was of high viscosity.

Torque produced by shearing of the fluid between the impeller and turbine is negligible for oils commonly used in fluid clutches, but it was found to be appreciable for the highly viscous test oil. Tests made

for various oil temperatures showed that temperature increases which reduce the viscosity result in a greater torque capacity because of the smaller friction losses. Tests using three different amounts of fluid showed that the quantity of fluid in the working circuit had a significant effect on the transmitted torque.

For a specified slip speed, torque was found to increase almost linearly as a function of input speed. For a given input speed, slip torque was found to be nearly linearly with respect to slip speed up to a value of 50 per cent slip. After this point the relationship was found to be unpredictable.

Tests were made to show the effect of clearance changes. Data were obtained for clearances of 1/32 and 1/16 inches greater than the nominal value. The changes resulted in only minor torque variations except for small fluid quantities.

Torque calculations are carried out using a cryogenic as the fluid. These calculations show that cryogenic fluids can be used to transmit substantial torque.

NOMENCLATURE

A	Area, ft^2; in^2
b	Radial width, inches
E	Voltage, volts
e	Ratio of turbine speed to impeller speed
F	Force, pounds
ΔF_c	Net centrifugal force, pounds
f	Dimensionless friction factor
g	Acceleration of gravity, 32.2 ft/sec^2
h	Head, ft
°K	Temperature, degrees Kelvin
L	Length, inches
M	Angular momentum, ft-lbs
N	Rotational speed, rpm
P	Power, hp; wetted perimeter, inches
Q	Heat flow, Btu/unit time
R	Resistance, ohms; ratio of flow area to wetted perimeter, inches
r	Radius, ft; in
S	Slip, per cent
s	Laplace operator
T	Torque, ft-lbs; in-lbs
t	Time, seconds
T°	Temperature, degrees Fahrenheit (°F)
$\bar{T}$	Temperature, degrees Rankine (°R)

NOMENCLATURE (continued)

V	Absolute velocity, ft/sec
V_c	Circulation velocity, ft/sec
V_r	Relative velocity, ft/sec
V_u	Tangential velocity, ft/sec
W	Flow rate, lbs/sec
w	Weight of fluid in impeller or turbine for steady state conditions, lbs
ε	Thermal emissivity
η	Efficiency, per cent
λ	Dimensionless coefficient
ν	Kinematic viscosity, centistokes
ξ	Dimensionless coefficient of friction
ρ	Density, lbs/ft^3
σ	Stefan-Boltzmann constant, 0.1714×10^{-8} Btu/hr-ft^2-°R^4
τ	Time constant, seconds
ω	Angular velocity, rad/sec

CHAPTER I

INTRODUCTION AND HISTORICAL BACKGROUND

Statement of Intent

Space exploration has required the usage of many systems employing clutches. Magnetic particle and disc clutches are commonly used, but the author could find no work being done on hydrodynamic fluid clutches for such uses. Therefore, the intent of this study is to investigate the characteristics, capabilities, and limitations of the hydrodynamic fluid clutch for possible applications in space.

Background

Power transmitters that use a fluid as the transmitting medium can be broadly classified as one of the following two types: 1) the positive displacement or hydrostatic type, and 2) the hydrokinetic or hydrodynamic type.

In the hydrostatic type, the power is transmitted by a positive displacement pump delivering fluid under high pressure and relatively low velocity to a positive displacement motor. The hydrostatic type will not be considered in this work.

In the hydrodynamic type, the power transmitted results from the change in the angular momentum of a fluid as the fluid flows radially outwards between the vanes of an impeller and radially inwards in a turbine.

Hydrodynamic power transmitters are further classified as torque converters and fluid couplings or fluid clutches. The torque converter is essentially a three element device. It has, imposed between an impeller on an input shaft and a turbine on an output shaft, a stationary member which produces a change in the torque between the impeller and turbine, resulting in a torque ratio between the two shafts. Thus, the converter contains the elements of a clutch but is, in addition, a transmission and will not be considered.

By using only two elements, an impeller on the input shaft and a turbine on the output shaft, which are separated from each other by a small clearance, a one to one torque ratio is obtained. This mechanically simple hydrodynamic transmitter is known as the fluid clutch.

When the steam turbine was introduced for marine propulsion early in this century, its output shaft had to be coupled directly to the propeller shaft because of the lack of a speed reducer. Since a steam turbine must run at high speeds to be efficient and a propeller at low speeds, the efficiency of the drive was low. This fact led to the development of the first hydrodynamic device, a torque converter, for the transmission of power. Interest in it was short lived because its 85-87 per cent efficiency could not compare with the much more efficient and cheaper helical gear reduction unit which was introduced around 1910. Although the fluid clutch was proposed for use with the helical reduction gear, no support was found for the idea.

With the appearance of the Diesel engine in the marine field, the fluid clutch was further developed by Dr. Bauer, director of the Vulcan Works of Hamburg, Germany, into what was called the "Vulcan Coupling".

The beginning of the fluid clutch in the industrial and automotive field began in 1929 when Harold Sinclair and the Associated Equipment Company of England accrued the world rights for industrial and automotive applications of the "Vulcan Coupling". Further designs were known as the "Vulcan-Sinclair Coupling".

The design of the fluid clutch has changed little since Sinclair, although its use in the automotive industry has resulted in a few alterations.

Literature Survey

The first pioneer work on the principle of hydrodynamic transmission was done by Froude, whose paper on a hydrodynamic dynamometer was read before the Institution of Mechanical Engineers in 1877 (1,2).

Dr. Föttinger (3,4), originator of the first hydrodynamic power transmission, filed patents describing both the hydrodynamic converter and clutch in 1905. From 1905 to 1915 he developed his torque converter and owned all international master patents until 1908.

In 1922 Dr. Bauer further developed Föttinger's fluid clutch. Extensive experimental tests were made on the clutch to determine the best geometry of the fluid working circuit, the optimum number of vanes, the efficiency, the characteristics with fluids of different density and viscosity, and the clutch's ability to damp out torsional vibrations (5). These tests showed that input torque equals output torque under all conditions of speed and slip; efficiency equals 100 minus the percentage slip under all conditions of torque, speed, and filling; and torque fluctuations on the output shaft were around 2 per cent of the input fluctuations.

Sinclair, in two of his papers (6,7), described his work on the fluid clutch for industrial and automotive purposes. His main contributions were the scoop tube clutch, the traction clutch, and the ring-value traction clutch. The traction clutch is a constant fill unit as no means are provided to vary the quantity of fluid in the clutch while in operation. Whereas, the scoop tube (or adjustable speed) clutch has means to control the fluid quantity at all times.

The use of the fluid clutch in the automotive industry in this country was pioneered by the Chrysler Corporation (8). Chrysler eliminated the core (9) and added an annular or half-torus ring at the center of the vanes of the turbine to overcome a critical vibration period (10). Borg and Beck (11) experimented on vane combinations to prevent an objectionable hum at critical slip speeds noticed in the development of their clutch.

As the traction type of fluid clutch has considerable drag torque (torque absorbed by the clutch when the output shaft is stalled), there have been various methods devised to reduce this torque. One such method which will be discussed later is the use of the anti-drag baffle. Another method uses steel blades that deflect when the torque load exceeds a preset value and prevents overheating of the fluid (12). To completely eliminate the drag torque, the clutch can be by-passed at low speeds (13).

In the adjustable speed fluid clutch, the quantity of fluid in the working circuit can be varied by leak-off ports. For rapid emptying of the circuit, dumping valves are used. Bellows type valves were originally used for this purpose but were later replaced by a flat diaphragm

(14). Rapid filling and emptying can also be achieved by using a spring loaded valve and the fluid pressure (15).

Considerable effort has been expended by companies in search of a suitable fluid. A mineral oil having a viscosity around 150 Saybolt Universal seconds is generally recommended for industrial applications (16). To get away from the problems of liquids, steel shot with a diameter of 0.024 inches are sometimes used with a modified circuit (17,18). This design has many of the advantages of the liquid type, plus an ability to obtain an efficiency of 100 per cent.

Although most fluid clutches have a maximum speed of 3600 rpm, special low horsepower units have been built for speeds as high as 15,000 to 20,000 rpm (19).

Numerous articles on the description, selection, and usage of the fluid clutch have appeared over the years in technical periodicals. Some of these articles appear in the Bibliography. Two recent books on the theory, design, and construction are by Wolf (20) and Kickbusch (21).

CHAPTER II

FLUID CLUTCHES

General Description

Figure 1 shows a typical section diagram of a fluid clutch with designating terminology indicated.

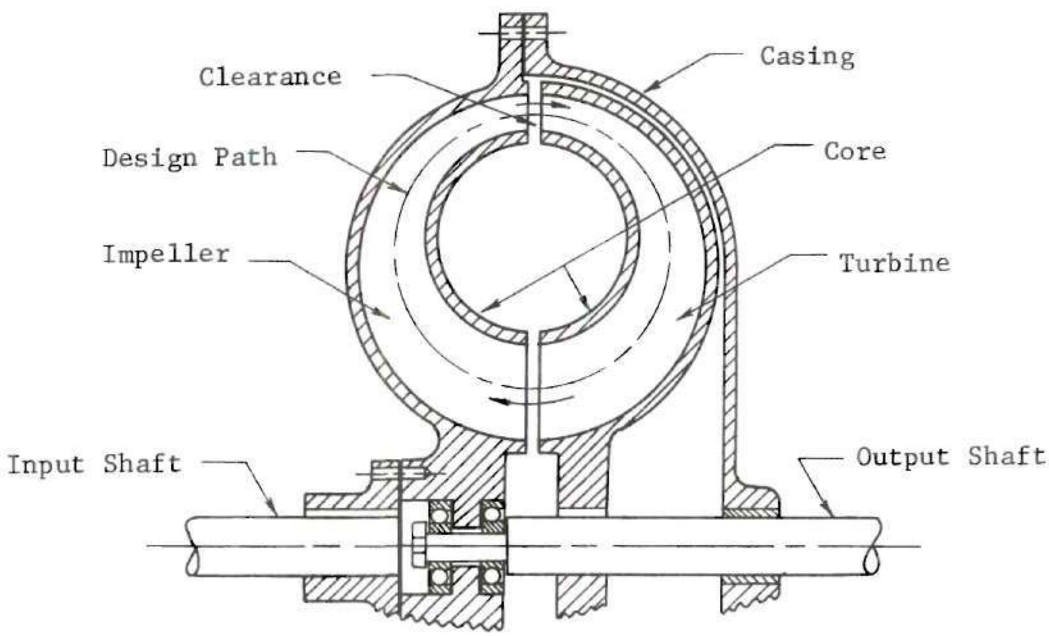


Figure 1. Sectional Diagram of Fluid Clutch.

The impeller, an integral part of one-half of the split housing, is connected to the input shaft. Vanes in the impeller are set radially to the drive shaft to guide the fluid and help prevent eddying. Opposite to the impeller is the turbine which is fixed to the output shaft and connected to the impeller only through the bearings. The major difference between the primary and secondary members is that the turbine usually has from 1 to 4 more radial vanes extending from the hub to the rim. If both the impeller and turbine had the same number of vanes, all of the vanes in one would pass the vanes in the opposite member at the same time which would cause fluid flow disturbances.

A small clearance or gap separates the impeller and turbine. This clearance and the fact that the turbine is not rigidly connected to the impeller leads to a considerable degree of flexibility in alignment and to a smooth vibrationless drive.

To retain the fluid, a casing or housing is bolted to the impeller, enclosing the turbine and connected to the output shaft by means of a rotating seal.

The annular core guide ring of semicircular section, one-half in the impeller and the other in the turbine, guides the fluid and reduces eddies. The eccentric location of this core keeps the flow area uniform for a constant fluid velocity.

Fluid clutches are generally constructed from welded steel, steel stampings, slotted and tabbed steel, cast steel, cast iron, or cast aluminum. For extremely high speeds, the impeller and turbine may be milled from a solid steel block. Cast aluminum is used to reduce inertia and weight.

The two main types of fluid clutches are the traction type and the variable speed type. Figure 1 is of the traction type. Since it is a constant fill clutch, it can not completely disengage the load from the driver and must, therefore, absorb the power when the load is stalled.

To obtain a varying volume of fluid within the circuit from 0 to 100 per cent and completely disengage the driver from the driven, the variable speed clutch incorporates (in addition to the impeller, turbine, and casing) an outer housing, leak-off ports, a scoop tube, an oil cooler, and generally a pump and reservoir (see Figure 16).

Principle of Operation

Referring back to Figure 1, assume the clutch to be full of fluid. As the impeller begins to rotate, centrifugal force on the fluid in the impeller causes the liquid to flow radially outwards between the vanes as indicated by the arrows. Crossing the clearance between the two members, it flows radially inwards in the turbine vanes until reaching the inner point of the circuit where it again crosses the clearance to the impeller, and the cycle is repeated. Fluid velocity in the circuit is known as the circulation velocity. Note that the turbine must turn at a slower speed than the impeller (generally 2-3 per cent slower) since the circulation velocity is created by the difference in the centrifugal pressures of the fluid between the primary and secondary members.

The above described circulation velocity and the rotational velocity of the two elements about the shafts axis causes the vortex ring of fluid to follow a path as shown in Figure 2.

Consider the unit mass of fluid, m_1, in Figure 2 at radius r_1. As m_1 flows with circulation velocity, V_c, to the position of unit mass, m_2, at radius r_2, its angular momentum increases until position 2. The difference in momentum at these two points is added to the impeller. From position 2 to position 1 the angular momentum of the unit mass decreases.

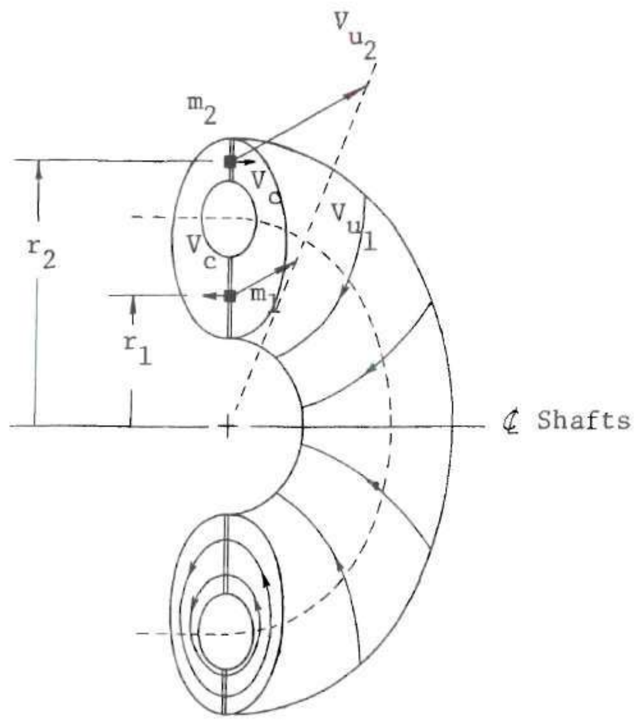


Figure 2. Rotating Vortex of Fluid.

Hence, for a clutch of a given diameter, the transmitted power is proportional to the density of the fluid and the circulation velocity.

For a clutch of a given diameter and a constant per cent slip (input speed, N, minus output speed divided by the input) the tangential velocities of the impeller at points 1 and 2, V_{u_1} and V_{u_2}, are proportional to the rotational speed, N. Consequently, the angular momentum change between points 1 and 2 in Figure 2 gives rise to the torque and is proportional to N. Since the slip was assumed constant, the mass flow rate, which is also a factor in torque, is proportional to the rotational speed. Thus, the power transmitted (torque times angular velocity) is proportional to the cube of the rotational speed.

Assume the rotational speed and the slip are constant. It follows that V_{u1} and V_{u2} are proportional to the diameter of the design path. The volume of fluid in the working circuit, which is proportional to the cube of the diameter, determines the mass of fluid. Therefore, the power or rate of doing work is proportional to the fifth power of the diameter. Since it determines the velocity changes and the mass, the diameter has the greatest effect on the transmitted power.

Theoretical Torque Transmission

The power that is transmitted through a fluid clutch is the results of three effects: change in the angular momentum of the circulating fluid, shearing of the fluid in the clearance between the impeller and turbine, and mechanical friction of the bearings. Fluid shear and bearing friction are normally negligible compared to the first effect. They are generally in the order of less than 1 per cent of the total power transmitted, depending on the type of bearings used and the viscosity of the fluid. Therefore, they will not be considered.

In the previous section a description of the principle of operation was given. Referring now to Figure 3, ω_I and ω_T are the impeller and turbine angular velocities, respectfully; r_1 and r_4 are the equivalent mean radii to the entrance area of the impeller and the exit area of the turbine, and they are equal; r_2 and r_3 are the equivalent mean radii to the impeller exit and turbine entrance annular areas, and $r_2 = r_3$.

Now, consider the exit section of the turbine. If the average absolute velocity of the fluid leaving the channel between two vanes

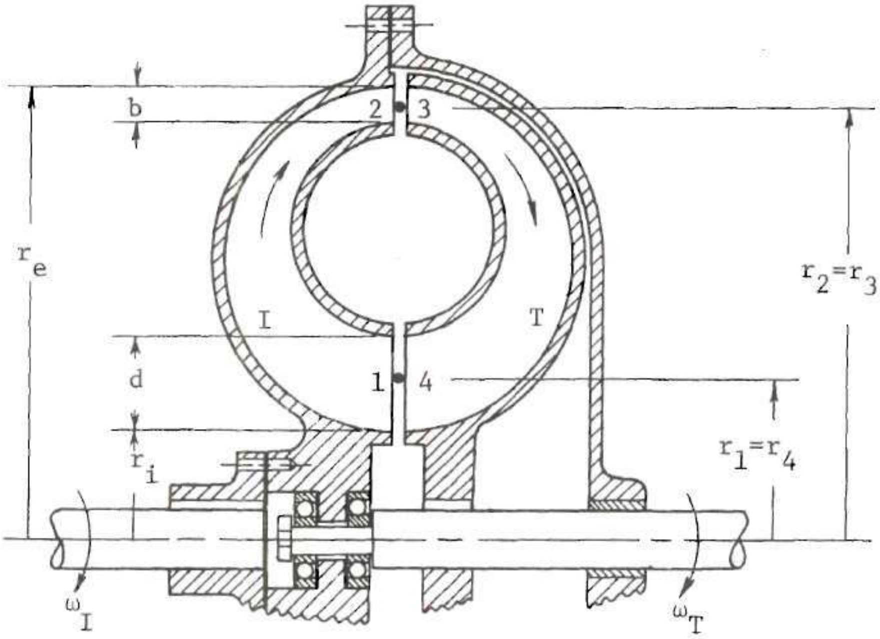
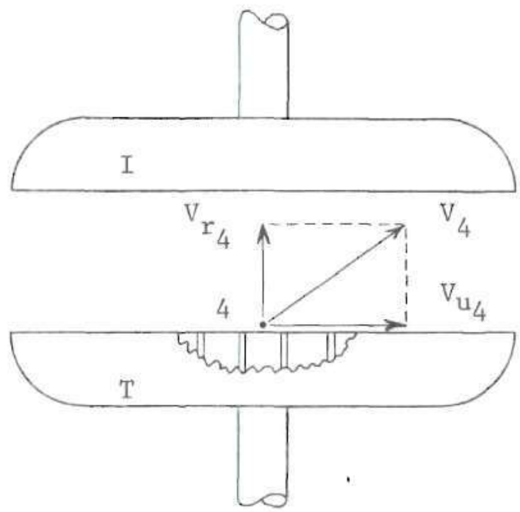


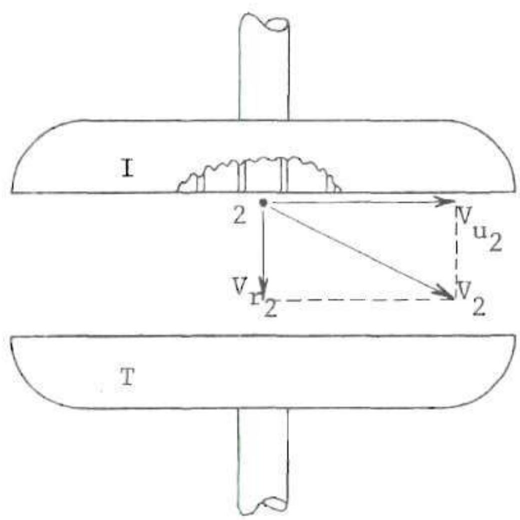
Figure 3. Traction Clutch Showing Notation.

in this section is taken at point 4, then this absolute velocity, V_4, is as shown in Figure 4(a). The circulation velocity of the fluid created by the difference in the centrifugal forces between the impeller and turbine gives the relative velocity, V_{r_4}, which is an axial component. The rotational velocity of the turbine wheel results in the tangential component, V_{u_4}. Note that the force associated with the axial component tends to separate the two members and has no moment around the shaft axis. Therefore, the angular momentum per unit time, M, of the fluid leaving the exit section of the turbine is

$$M_4 = \frac{W_4}{g} r_4 V_{u_4} \qquad (2.1)$$



(a)



(b)

Figure 4. Fluid Velocities Leaving the Turbine and Impeller.

where W_4 is the weight flow rate from the turbine. This angular momentum is also the angular momentum of the fluid per unit time entering the impeller, i.e.,

$$M_1 = M_4 \tag{2.2}$$

In a like manner, the angular momentum per unit time of the fluid leaving the impeller is

$$M_2 = \frac{W_2}{g} r_2 V_{u_2} \tag{2.3}$$

which must be the same as that entering the turbine.

For steady-state conditions the flow rates must be equal, i.e., $W_1 = W_2 = W_3 = W_4 = W$. The total change in the angular momentum about the axis must also equal zero under steady state. Since torque equals the change in the angular momentum with respect to time,

$$T_I + T_T = \frac{dM}{dt} = 0 \tag{2.4}$$

where T_I is the torque on the impeller resulting from the difference in the angular momentum of the fluid leaving and entering the impeller as given by

$$T_I = \frac{W}{g} (r_2 V_{u_2} - r_4 V_{u_4}) \tag{2.5}$$

The torque on the turbine, T_T, results from the angular momentum change of the fluid entering and leaving this member and is given by the equation

$$T_T = \frac{W}{g}\,(r_2 V_{u_2} - r_4 V_{u_4}) \qquad (2.6)$$

From Equation (2.4) the impeller and turbine torques are equal and opposite. Noting that $r_1 = r_4$, $r_2 = r_3$, and $V = r\omega$, the theoretical torque transmitted through the clutch is

$$T = \frac{W}{g}\,(r_2^{\,2}\omega_I - r_1^{\,2}\omega_T) \qquad (2.7)$$

CHAPTER III

THEORETICAL TREATMENT OF TEST CLUTCH

In the preceding chapter the theoretical torque equation was derived using a clutch with a core and vanes that extended from the inner most radius from the axis to the outer radius. The clutch used for the experimental tests was not of this construction as can be seen in Figure 5, but the general principle and torque equation in Chapter II applies. No core was present and the vanes (16 in the impeller and 18 in the turbine) were quadrant shaped. Dimensions shown in Figure 5 were measured from the clutch.

If, when the clutch is in operation, the fluid circulates in a circular shape as shown in Figure 6, the quantity of fluid (neglecting the vanes) in the working circuit is the volume of a torus. Referring to Figure 7 and using the measured dimensions of the clutch, this volume is about 25.2 in^3. The width of a vane averaged around 0.2 inches, so after subtracting the volume of the torus occupied by the vanes, the actual volume of fluid in the circuit was 21.7 in^3. Using the fluid whose density was 56 lbs/ft^3, the weight of fluid is 0.704 lbs, which compares favorably with the 0.7 lbs used in the tests.

Under steady-state operation, the rate of flow will be the same in the impeller and turbine. For a constant circulation velocity, the cross-sectional area of flow must be the same. Since the channels between the vanes become wider,

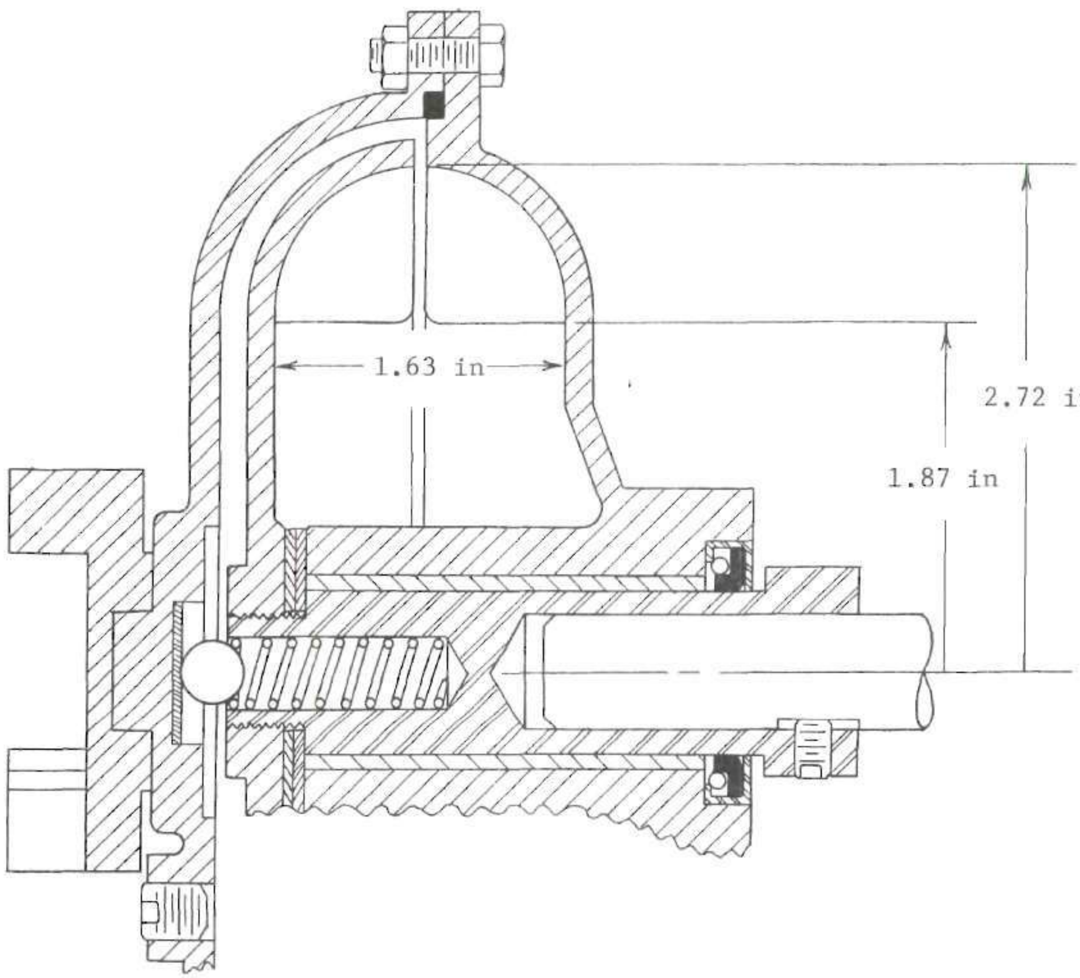


Figure 5. Diagram of the Test Clutch.

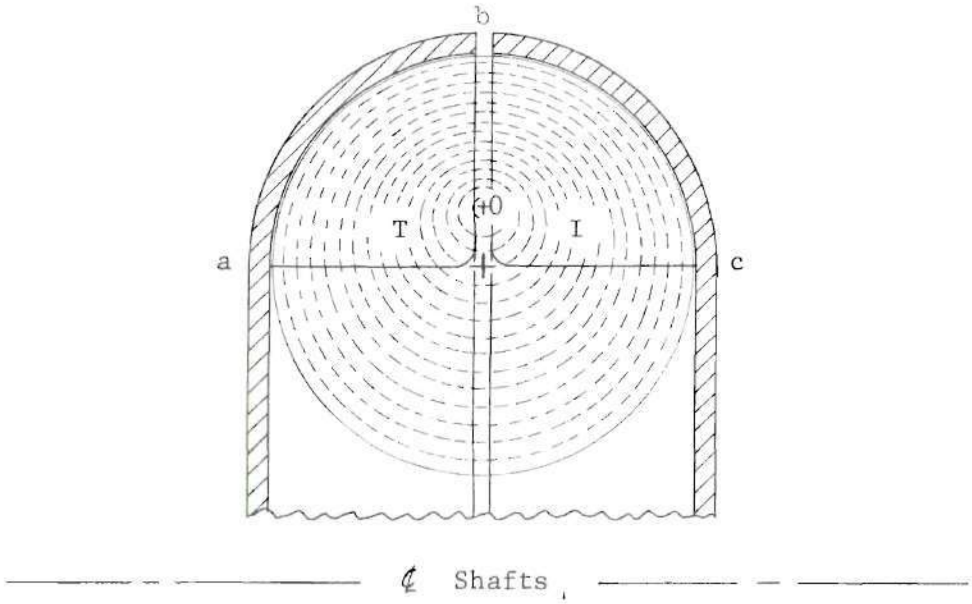


Figure 6. Theoretical Shape of Fluid in Operation.

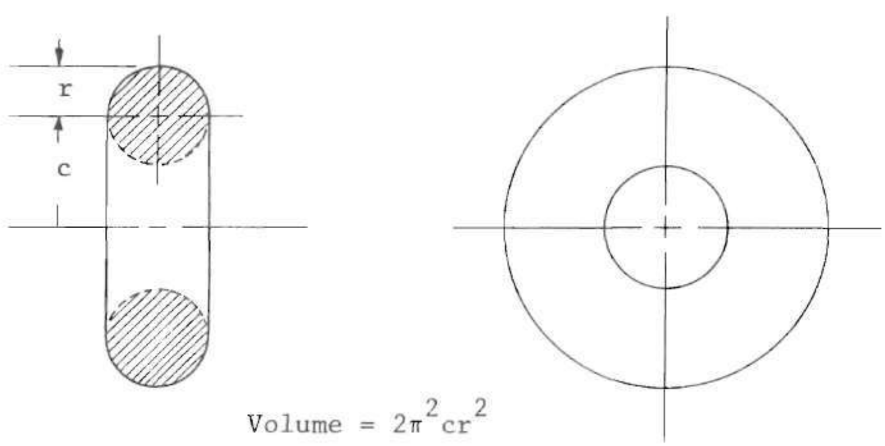


Figure 7. Schematic of Torus.

circumferentially, with distance from the axis, the fluid must circulate around some point, such as point 0 in Figure 6, to insure a uniform velocity. Note that point 0 is not the center of the circular shape.

This point around which flow takes place can be determined with the aid of Figure 8. In this figure r_e = outside radius, r_i = inside

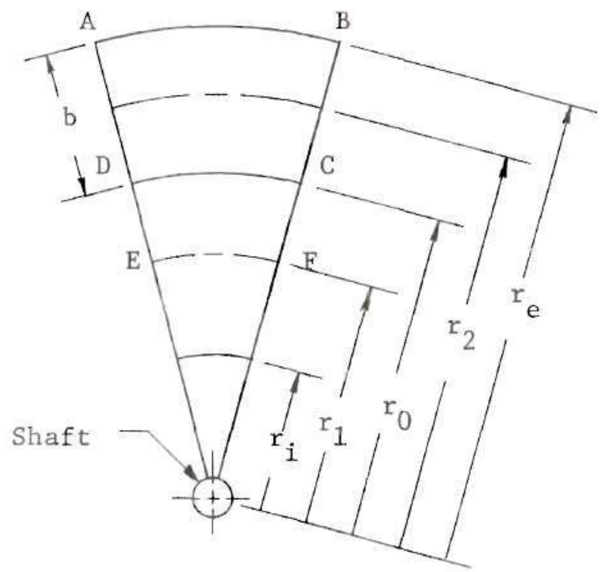


Figure 8. Front View of Channel Between Two Vanes.

radius to fluid torus, and r_o = radius to the center of rotation of the circulating fluid. Now, the fluid enters the turbine (leaves the impeller) through area ABCD and flows out of area CDEF. These areas must be equal, so by equating them to each other, the radius, r_o, to the axis around which the fluid flows is

$$r_o = \sqrt{\frac{r_i^2 + r_e^2}{2}} \tag{3.1}$$

Since some of the fluid flows along the outer walls of the impeller and turbine, its velocity is greater than that portion of fluid that flows along the short path around the axis of circulation (axis located at radius r_o in Figure 8). It is therefore desirable to use an average velocity and effective radii which will produce the same transmitted torque as flow through the actual circuit.

Let r_1 (see Figure 8) be the equivalent mean radius to the entrance area of flow of the impeller (exit area of turbine) at which point the angular momentum per unit time can be assumed to be concentrated. Let r_2 be the equivalent mean radius of outflow to the impeller (entrance to turbine). These radii were derived by Heldt (22) to be approximately

$$r_1 = \sqrt[3]{\frac{r_o^4 - r_i^4}{4\ (r_o - r_i)}} \tag{3.2}$$

and

$$r_2 = \sqrt[3]{\frac{r_e^4 - r_o^4}{4\ (r_e - r_o)}} \tag{3.3}$$

Substituting values in Equations (3.1), (3.2), and (3.3), $r_o = 2.072$ inches, $r_1 = 1.631$ inches, and $r_2 = 2.420$ inches.

In order to calculate the torque transmitted, the pounds of fluid circulating between the primary and secondary elements must be known. This flow rate can be found if an effective circulation velocity is determined. Since this velocity is dependent on the frictional resistance which is in turn dependent on the velocity, a solution is possible if relations between the frictional loss factor and the circulation velocity are known. Bruckner (23) used the Darcy-Weisback pipe flow equation and a friction factor chart to determine the velocity by a trial and error type solution, but his friction factors were too low for the clutch tested. Therefore, an equation for the friction factor will be derived using the Chézy formula for turbulent flow in open and closed conduits. Chézy's formula (24) is as follows:

$$\frac{h_f}{L} = \frac{\lambda}{R}\frac{V_c^{\,2}}{2g} \tag{3.4a}$$

where

h_f = Head loss due to friction
L = Characteristic length
λ = Dimensionless coefficient
V_c = Fluid velocity
R = Ratio of area to wetted perimeter of the conduit

The purpose in deriving an equation for the friction factor is to be able to predict the transmitted torque for fluids other than the oil used in the tests. Characteristic dimensions that will be used in the derivation were somewhat arbitrarily selected, but their selection is justified because they affect only the coefficient, λ, and are independent of the fluid. The derived equation should then be valid for any Newtonian fluid used in the test clutch.

Now, visualize the half of the torus of fluid (abc in Figure 6) rubbing on the walls of the circuit unwound into a trough shape, and then straightened out into a rectangular slug shape. Take the characteristic length, L, of the Chézy equation equal to the length abc or one-half the circumference, $2\Pi(r_e - r_o)$ (see Figure 8). Let the wetted perimeter, P, equal the circumference, $2\Pi r_2$. Note that this perimeter was arbitrarily selected but it should be a characteristic of the clutch and not vary with fluid. Let the area, A, be the area of flow, $\Pi(r_e^2 - r_o^2)$. If λ is taken as f/4, where f is a friction factor, then the Chézy formula becomes

$$\frac{h_f}{L} = \frac{f\ V_c^2}{8g\ \frac{A}{P}} \qquad (3.4b)$$

Solving for f,

$$f = \frac{8g\ h_f\ A/P}{LV_c^2} \qquad (3.4c)$$

Substituting known values for the terms of Equation (3.4c),

$$f = 81\ \frac{h_f}{V_c^2} \qquad (3.4d)$$

By using Equations (2.7) and (3.4d), the test data obtained for the less viscous oil used in the tests at an input speed of 1800 rpm, and a modification of a friction factor equation given by Huebotter (25), an empirical equation for the friction factor results as follows:

$$f = 15 \left[\frac{v}{bV_c}\right]^{1/4} \qquad (3.5)$$

where

v = Kinematic viscosity, centistokes

b = Radial width (see Figure 8), inches

V_c = Circulation velocity, $\frac{ft}{sec}$

The net centrifugal force on the fluid between the impeller and turbine gives rise to the circulation flow. This net force, ΔF_c, in pounds can be derived (23) and is

$$\Delta F_c = \frac{w}{g} \pi^2 \frac{r_o}{3} \left[\left(\frac{N_I}{60}\right)^2 - \left(\frac{N_T}{60}\right)^2\right] \qquad (3.6)$$

where

w = Weight of fluid in the impeller or turbine, lbs

r_o = *Radius (see Figure 8), inches*

N_I = Impeller rpm

N_T = Turbine rpm

The net centrifugal head, hc, in feet is then

$$hc = \frac{\Delta F_c}{A\rho} \qquad (3.7)$$

where

A = Area through which the fluid flows, ft^2

ρ = Density of fluid, lbs/ft^3

Since the fluid flows in a closed circuit, it is retarded only by the resistance due to friction. Hence, the net centrifugal head must equal the frictional head loss in Equation (3.4d), i.e.,

$$hc = h_f \tag{3.8}$$

For any Newtonian fluid, Equations (3.4d), (3.5), (3.6), (3.8) and the torque equation (2.7) (in which $W = V_c A\rho$) can be used to determine the transmitted torque for the test clutch.

CHAPTER IV

EXPERIMENTAL INVESTIGATIONS

Experimental Equipment

The arrangement of experimental equipment used is shown in the block diagram of Figure 9. A ten foot lathe bed was used to mount the test apparatus from the servo valve to the load pump.

Section I of Figure 9 is the drive section. A hydraulic drive was used because of its stiffness and speed range capabilities.

Speed control (Section II) was obtained by a velocity control feedback system. The input and output speed of the clutch was read from an electronic counter.

Section III is the torque measuring section. The clutch was loaded by the hydraulic pump. Torque from the strain gage torque transducer was read from an oscilloscope.

A detailed description of the instrumentation used is given in Appendix A.

Experimental Procedure

To accomplish the objective of this work a typical, small horsepower, traction fluid clutch was tested. A few of the factors that influence the performance of this clutch are: 1) the amount of fluid in the clutch, 2) the input speed, 3) the difference between the input and output speeds (slip), 4) the temperature and viscosity of the fluid, and 5) the circuit design. Therefore, tests were based on determining the

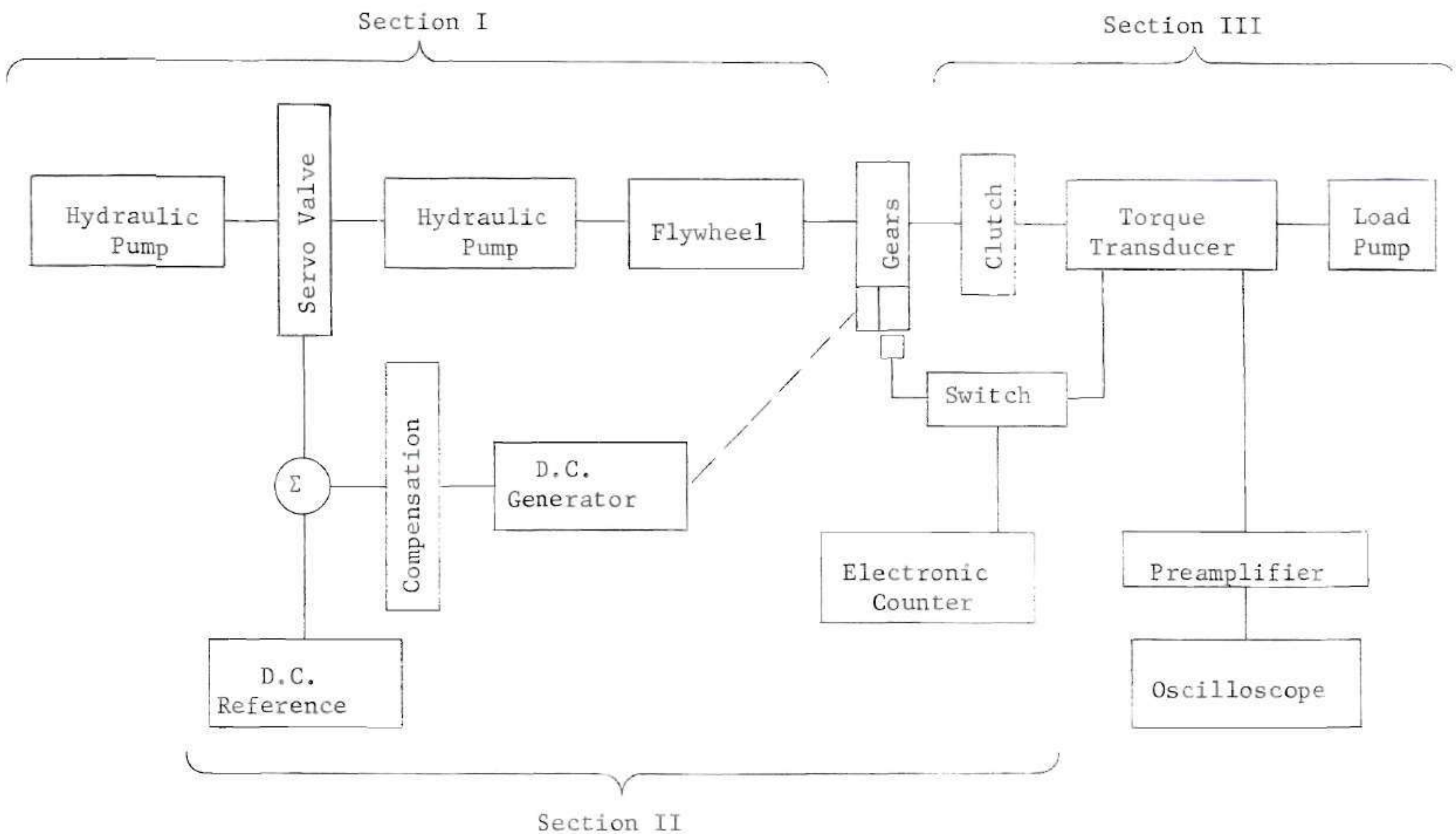


Figure 9. Block Diagram of the Experimental Apparatus

effects of each factor

Two different fluids were used in the tests -- a light weight oil with a measured density of 56 lbs/ft^3 and a high viscous oil with a density of 55 lbs/ft^3. After testing, the viscosity of the oils was measured by a Saybolt Universal viscometer at the standard temperatures, and the results are shown in Table 1.

Table 1. Viscosity of Oils

Oil	Temperature °F	Viscosity S.U.S.
	100	190
Light Weight	130	105
	210	48
	100	2390
Heavy Weight	130	900
	210	135

The quantity of oil in the clutch was changed. Tests were run using 5 oz, 8 oz, and 11.2 oz of both the light and heavy weight oil.

Drag torque, which is the transmitted torque when the output is fixed or stationary, was measured as the clutch input speed was varied from 300 rpm to 1800 rpm in increments of 100 rpm. Drag torque tests were conducted for oil temperatures of 100°F±5°, 125°F±5°, 150°F±5°, and 175°F±5°. Temperatures were taken after the run for each input speed setting. Since viscosity is a function of temperature, this test shows the effect that viscosity has on the torque transmitted through the clutch.

To determine the effect of slip on the performance, the torque on the clutch was increased by restricting the flow from the load pump as described in Appendix A. The input speed to the clutch was held constant and the output speed of the clutch was read from an electronic counter.

Although the geometry of the clutch could not be changed, the gap separating the impeller and turbine was increased $\frac{1}{32}$ inches and $\frac{1}{16}$ inches from its initial value (which was approximately $\frac{5}{64}$ inches) by shims, and drag torque and torque-slip tests were made.

Discussion of Results

Degree of Fill

An advantage of the fluid clutch is its ability to limit the stalling torque and, thereby, protect the driver. As no means are provided to vary the fluid in a traction clutch while it is in operation, the point at which it will stall depends on the degree of fill. Figure 10 shows typical curves.

The more fluid in the working circuit, the larger the size of the power transmitting vortex and, hence, greater torque.

Torque-Slip Relations

Figure 11 shows a typical experimental torque-slip plot for the clutch tested at an input speed of 1800 rpm. In Chapter II it was observed that the torque transmitted was proportional to the circulation velocity of the fluid. The difference in the centrifugal force acting on the mass of fluid in the impeller and the force acting on the liquid in the turbine create this circulation velocity. Therefore, the circulation velocity is proportional to the difference of the squares of the angular velocities. Resistance to the

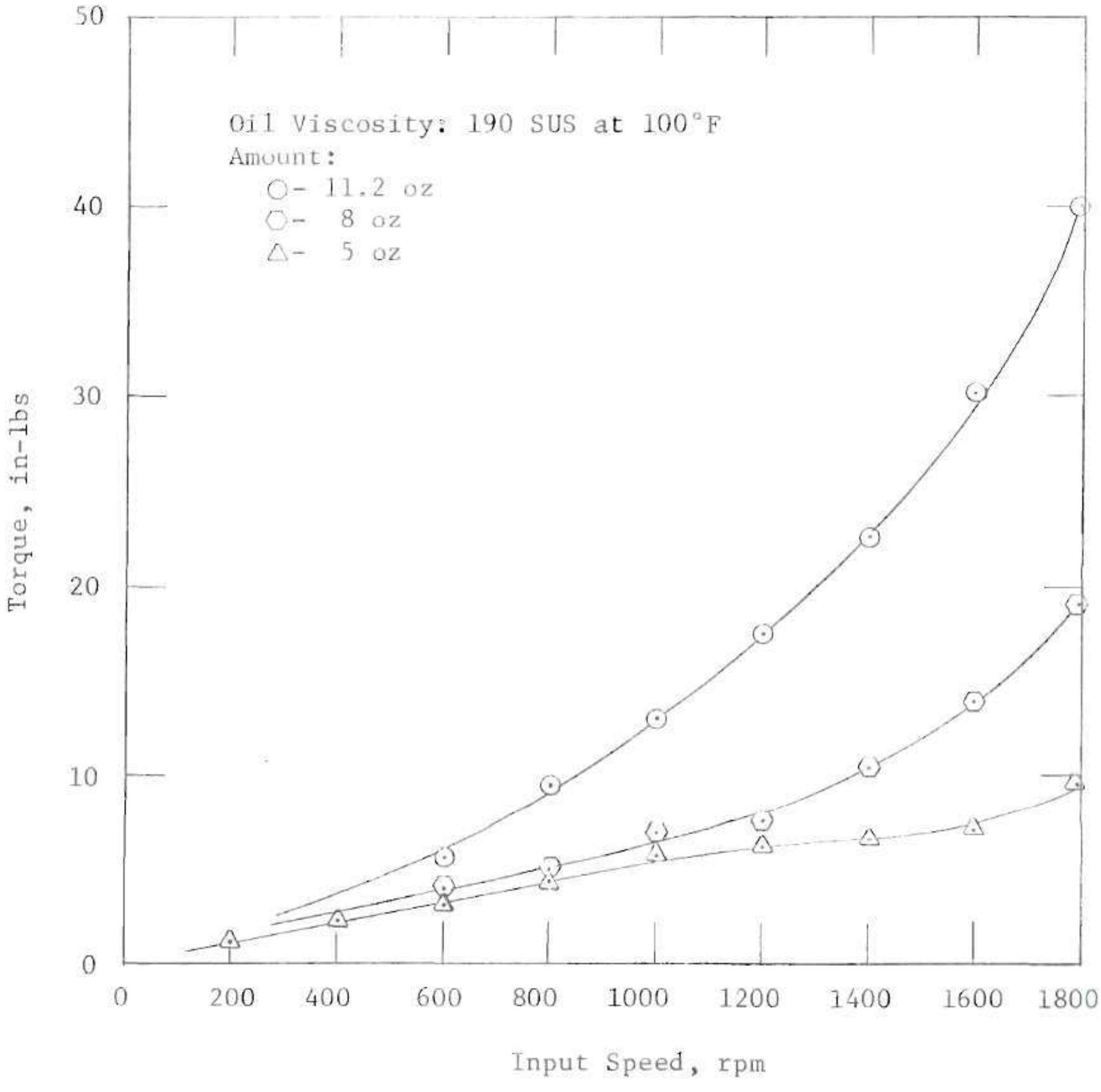


Figure 10. Drag Torque for Various Degrees of Fill.

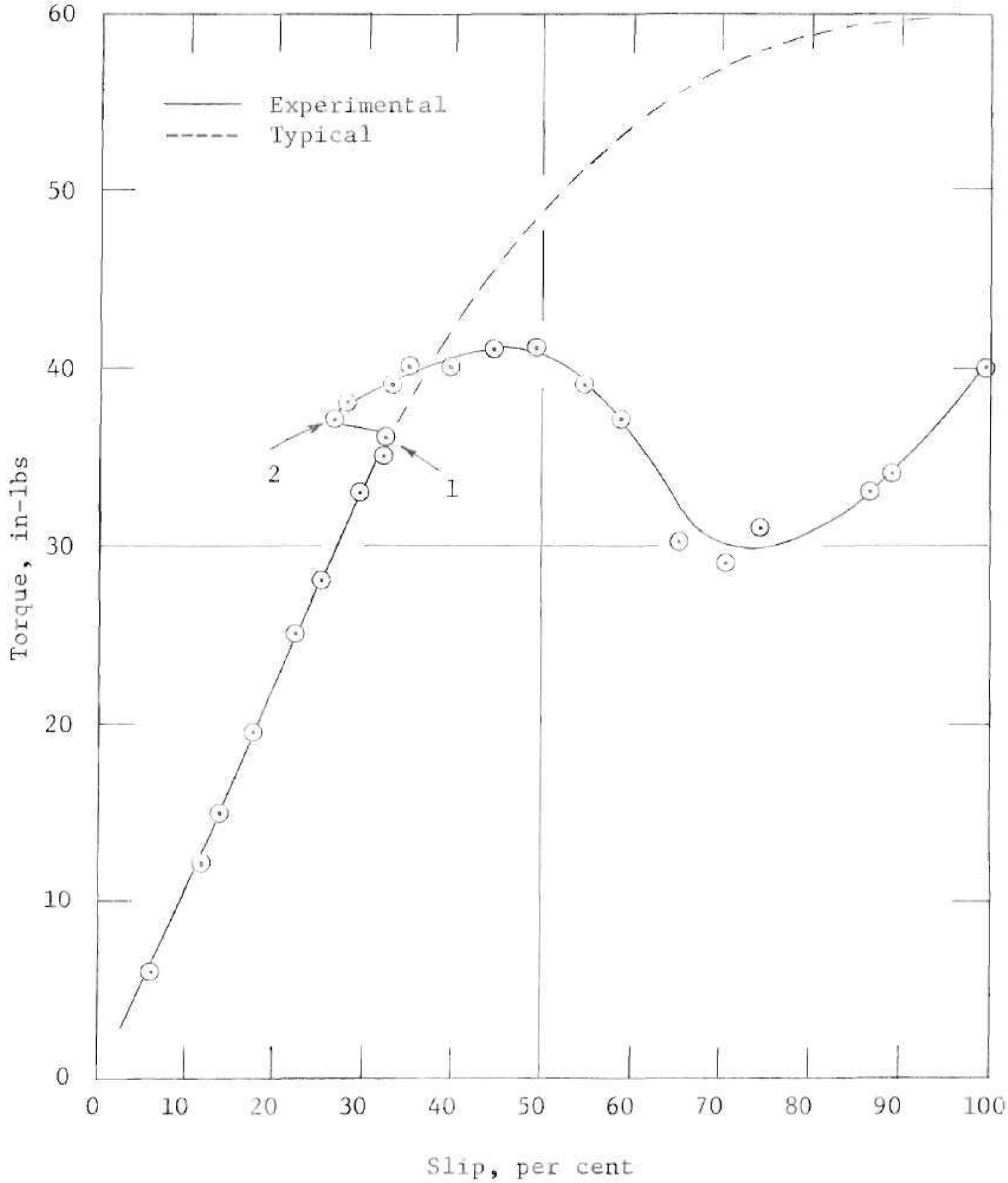


Figure 11. Typical Torque-Slip Relations

circulation flow is due to friction, which varies as the square of the circulation velocity. Hence, the circulation velocity and the transmitted torque should be proportional to the per cent slip with both increasing as slip increases. A typical theoretical curve is shown by dashed lines in Figure 11.

The experimental curve reaches a maximum torque value at approximately 50 per cent slip and then follows an unpredictable path at a torque less than maximum. It is thought that this decrease in torque is due to the extremely high eddies at higher slip values, which prevents the formation of defined vortex flow.

The sudden decrease in slip with little change in torque noticed in the experimental curve of Figure 11 occurred at the same clutch output speed for all conditions of filling and for both oils used. As torque was applied to the clutch, slip increased in an approximate straight line relation until reaching 32 per cent slip (1225 rpm output speed in this case). At this point the slip decreased to 26 per cent (output speed of 1332 rpm). It was first thought that this deviation was caused by an unstable condition in the load hydraulic pump or by a critical frequency of the output shafting, since it was a function only of the output speed and did not depend on the input speed or degree of fill. Also, large torque fluctuations were noticed on the oscilloscope in this slip range.

To check the hydraulic pump and shafting as possible causes, tests were made with and without the clutch, and the results displayed on the storage oscilloscope and recorded with a Polaroid camera. Figures 12(a) and (b) show the results for speeds of 1400 rpm and 1275 rpm without the

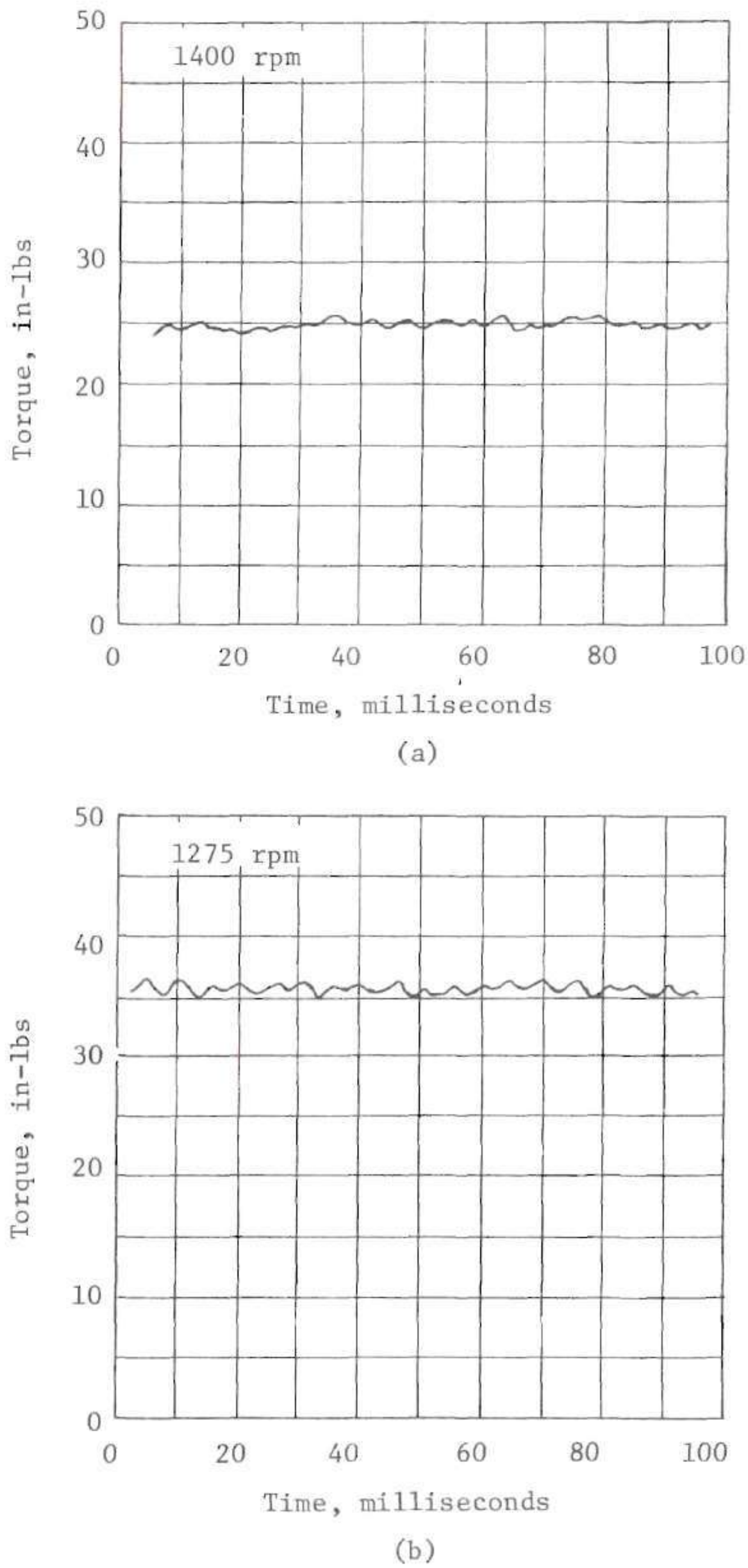


Figure 12. Torque Fluctuations Without Clutch.

clutch, and 13(a) and (b) are the results with the clutch at output speeds of 1400 rpm and 1275 rpm. The 1275 rpm run using the clutch was recorded after the slip had reached 32 per cent (point 1 in Figure 11) but before it had decreased to 26 per cent (point 2). Input speed to the clutch was 1800 rpm in both cases.

It therefore appears that the period of instability was caused by the clutch.

Temperature Effects

The operating temperature of the oil can have a great deal of effect on the delivered torque as can be seen from the typical experimental drag torque curves of Figure 14. In this figure temperature, T_2°, is approximately 75 per cent greater than T_1°. The torque increase due to the higher oil temperature was around 25 per cent at an input speed of 1800 rpm.

The rise in temperature lowers the viscosity and density of the oil, but the torque increases because of the greater flow circulation resulting from the reduced resistance to flow. How much the torque will increase depends on the effect temperature has on the viscosity and density of the oil.

Viscosity

For two liquids with the same density and at the same temperature, the less viscous fluid should theoretically result in more torque. However, this was not the general result observed. In the tests, the one with the higher viscosity generally resulted in higher torque at an oil temperature of 100°F.

Its high viscosity at 100°F, 2390 SUS, apparently produced a large

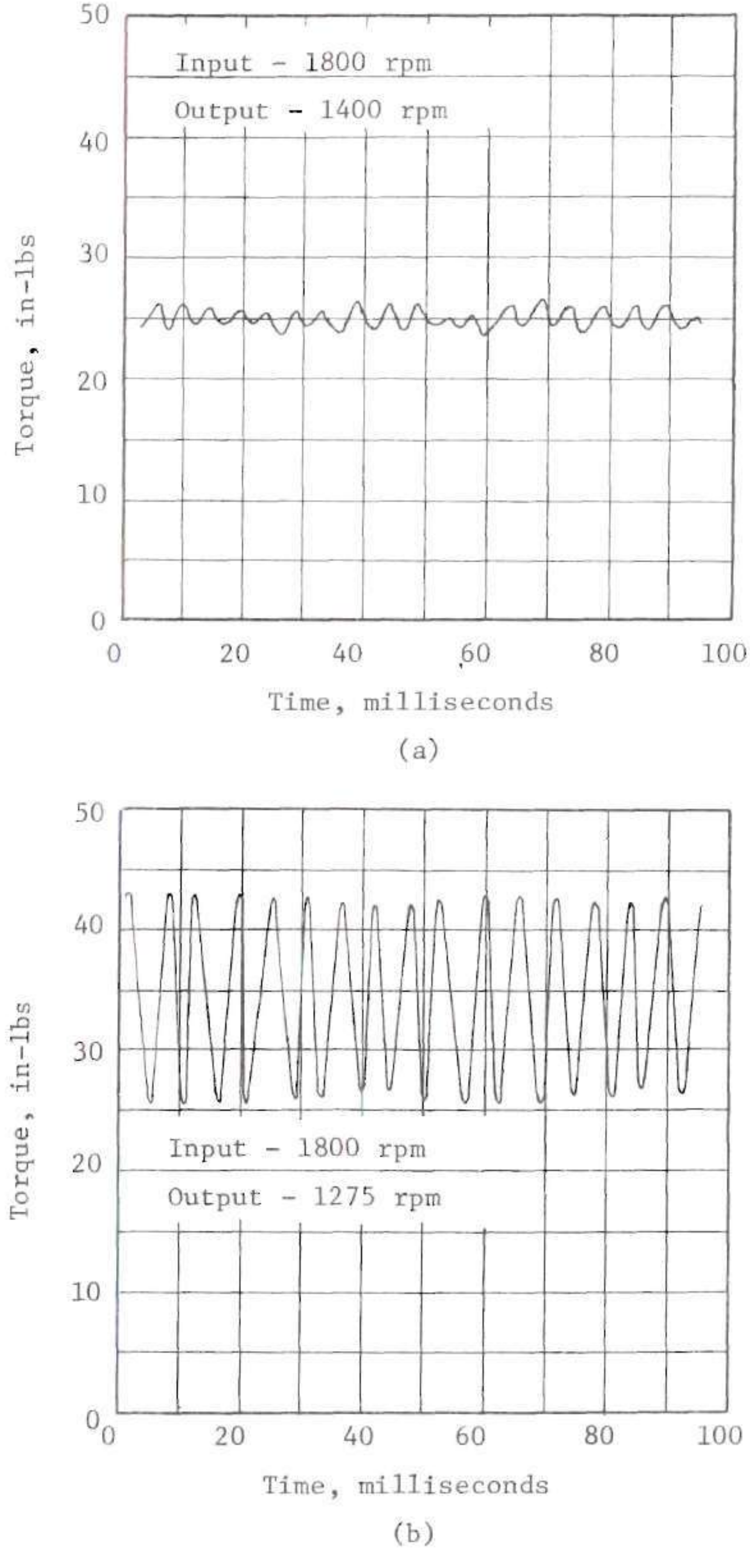


Figure 13. Torque Fluctuations With Clutch.

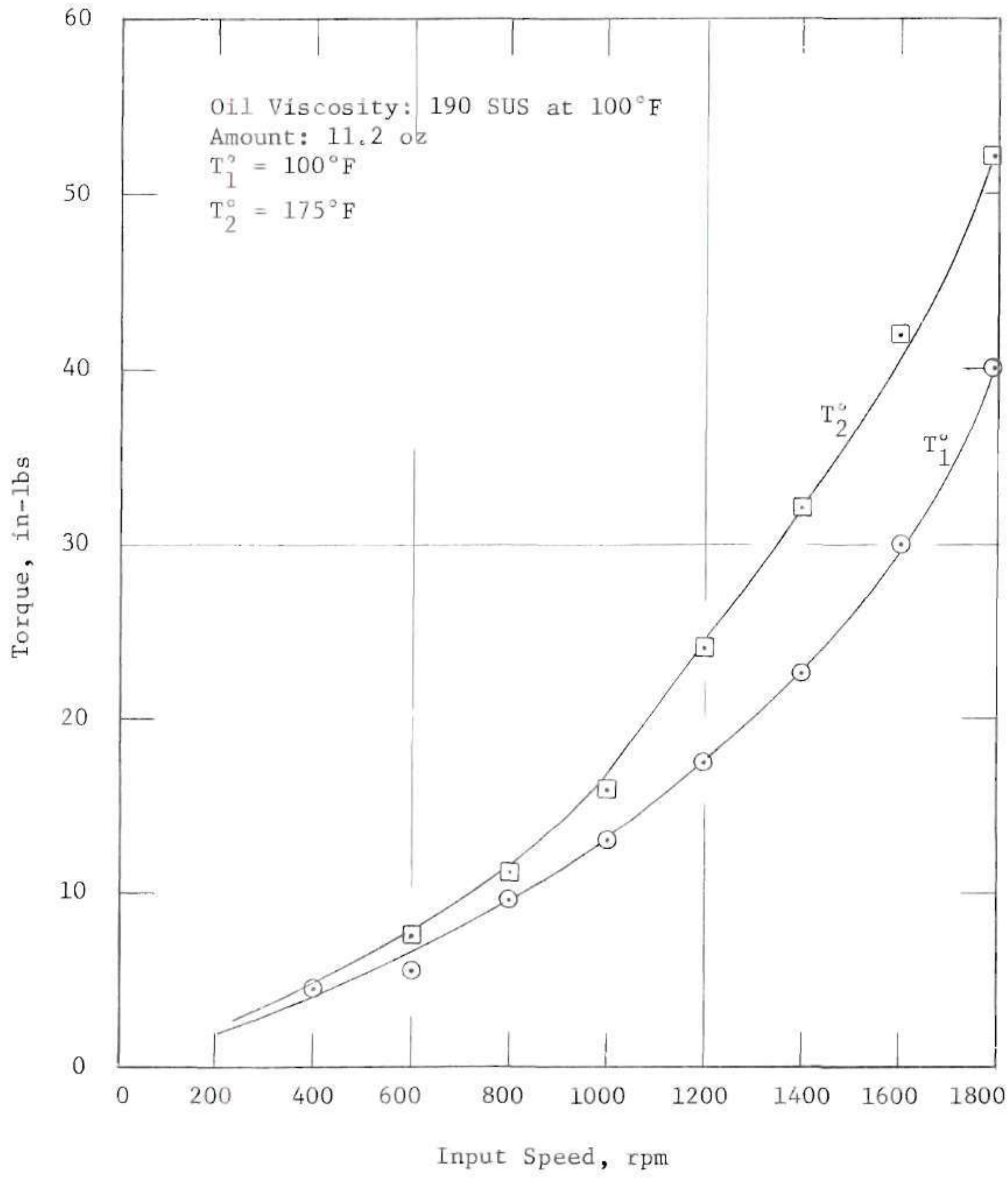


Figure 14. Drag Torque Curves Showing Temperature Effects.

shear torque. As the temperature of the oils increased, the shear torque decreased and the less viscous oil produced more torque as can be seen in Appendix B.

Clearance Changes

The gap or clearance between the impeller and turbine was increased by $\frac{1}{32}$ inches and $\frac{1}{16}$ inches. No significant change in transmitted torque was observed over the torque produced by the standard clearance, except for torque-slip tests using 5 ounces of oil. In these tests the torque was less for the greater clearances.

CHAPTER V

DESIGN CONSIDERATIONS

Circuit Designs

Traction Clutch

Various circuit designs have been devised for the fluid clutch to obtain desired characteristics. Some of these designs are shown from (a) through (f) in Figure 15, taken from Reference (26).

A normal traction clutch with radial vanes and a true torus shape circuit is shown in (a). To help prevent eddying and obtain better control of the fluid flow, a core can be added (b).

Since the traction clutch can not completely disconnect the driver from the driven, it is desirable to minimize the transmitted torque under certain conditions. The anti-drag baffle in (b) reduces the transmitted torque at low output speeds and under starting conditions. This baffle operates in the following way. When the impeller is turning at a much higher speed than the turbine, the centrifugal force on the fluid in the turbine is low. A high circulation velocity of the fluid results from the large unbalanced centrifugal forces, so the fluid clings to the boundary of the circuit and is broken up by the baffle. Transmitted torque is thereby reduced. As the turbine speed increases, the fluid vortex draws away from the baffle because of the increasing centrifugal force.

A deaerating chamber (c) is connected to the center of the core

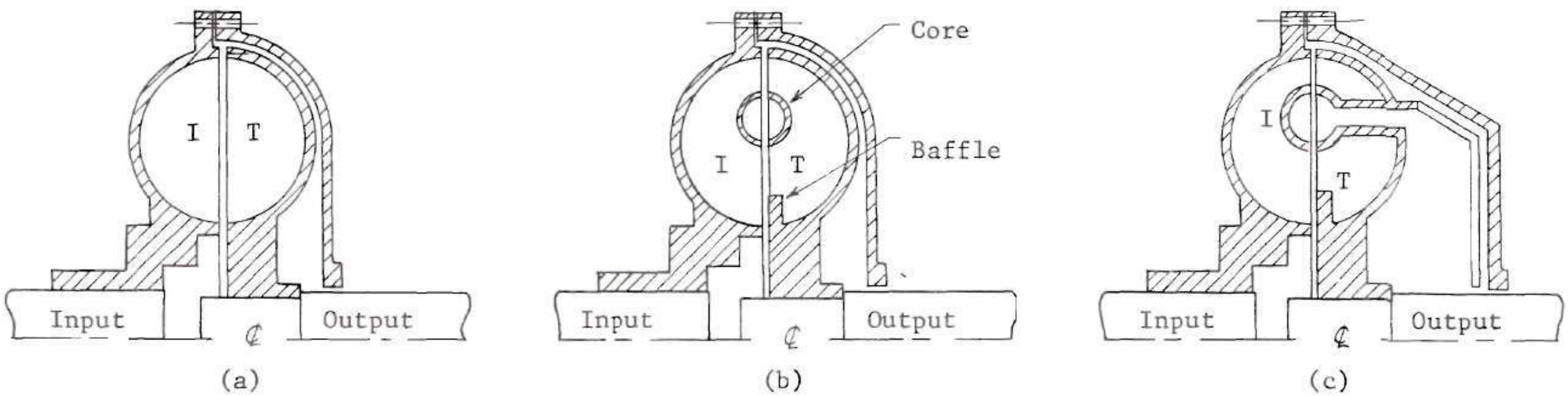


Figure 15. Circuit Designs.

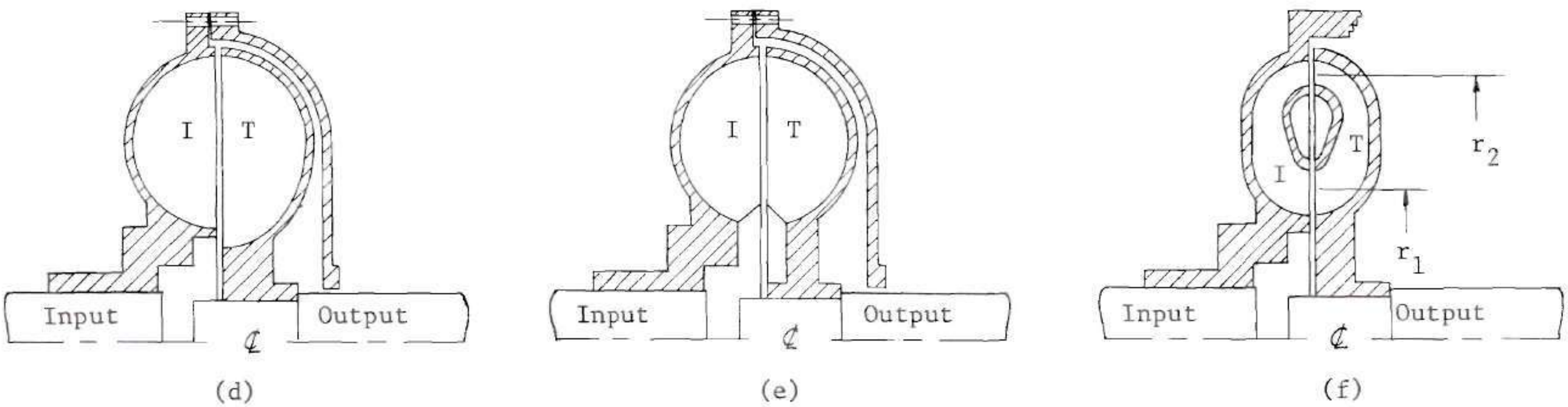


Figure 15. (continued).

to dump part of the fluid from the working circuit and to mix the displaced air into the fluid, thereby reducing the drag torque when the load is stalled. The air separates from the oil and the dumped oil returns to the working circuit when the turbine starts to rotate.

Another modification to reduce the drag torque is to use an eccentric runner (d). With the turbine stalled, the eccentric runner serves the same purpose as the anti-drag baffle. Another method used in reducing the drag torque on large sizes can be seen in (e). This torque reduction is produced by the opening left in the inner part of the circuit.

By increasing the size of the core in the shape shown in (f), the torque capacity is increased under normal running conditions, but the drag torque is reduced. The capacity increases because the mean radius, r_1, at which the fluid enters the impeller is reduced and the mean radius, r_2, at which the fluid leaves the impeller is increased. An increase in the angular momentum results. Drag torque reduction stems from the greater friction loss due to the reduced total area. This modification is used in a twin-type clutch design.

Adjustable Speed Clutch

In addition to duplicating functions of the traction clutch in protecting the load from shock, vibration, and excessive torque, the adjustable speed clutch offers infinitely variable speed regulation, drive equipment starting under no load, load acceleration control from seconds to minutes, and complete disengagement of the load from the driver.

There are three common basic arrangements of the adjustable speed clutch. These are shown in Figure 16. The first arrangement (a) has the same basic parts as the traction type but, in addition, has the rotating reservoir and an adjustable scoop tube. As the clutch is put into operation, the fluid bleeds off through the leak-off ports into the rotating reservoir. By adjustment of the eccentrically swivelled scoop tube, the amount of liquid in the working circuit can be controlled. The velocity of the fluid entering the scoop tube is high enough to carry it through the oil cooler and back into the circuit.

To prevent overheating of the fluid in the traction clutch, it depends on convective heat transfer. Air impeller blades are normally attached to the casing to help in cooling. In the adjustable speed type the outer housing severely reduces the convective heat transfer, and during high slip operation when the vortex is small, excessive heating would result if the clutch relied on external cooling. Therefore, an oil cooler is generally necessary.

Dumping valves can be used in place of the leak-off ports to permit rapid emptying of the circuit. A few types are the bellows, flat diaphragm, ring, piston, and spring loaded. Declutching can be achieved from one second and up with dumping valves, depending on the size and load.

In arrangement (b) there is a fixed reservoir and a one direction circulation pump used in conjunction with the adjustable scoop tube. The pump sends a constant volume of liquid through the external cooler and into the working circuit. Approximately 1/4 of 1 per cent of the input power is required by this pump. Fluid enters the scoop tube as in the first arrangement, but in this case it goes into the reservoir.

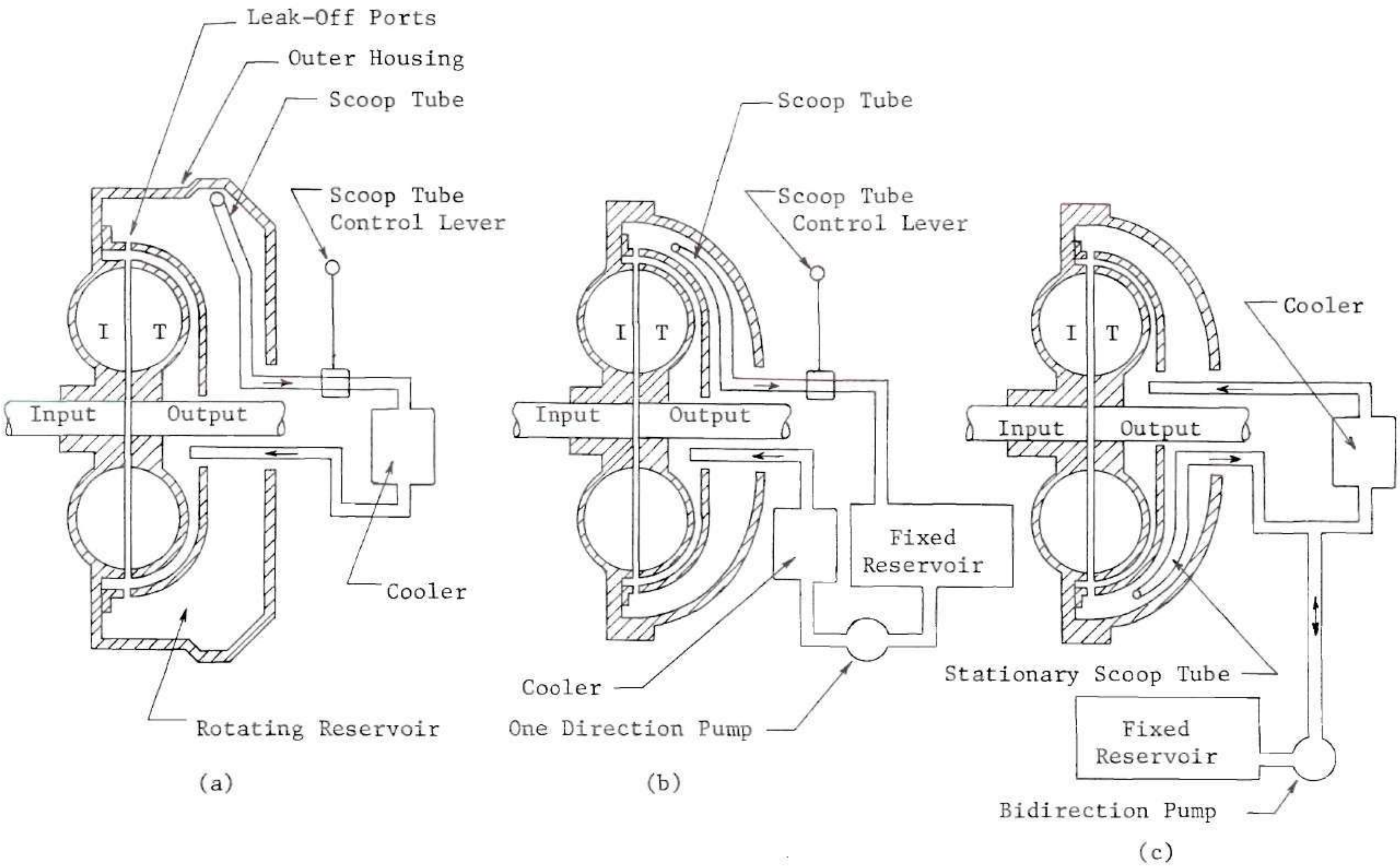


Figure 16. Basic Arrangements of Adjustable Speed Clutch.

The reservoir stores part of the fluid under starting conditions, provides an expansion chamber, and separates air from the fluid.

The last arrangement (c) differs from the second by having a stationary scoop tube and a bidirectional circulation pump. In this design the scoop tube serves only as a means of withdrawing the fluid from the housing. The size of the vortex being determined by the bidirectional pump. It runs in one direction for declutching and in the opposite for engagement. Continuous circulation of the fluid through the cooler occurs when the pump is off.

Selection Tables

In Tables 2 and 3 are shown typical selection data for traction and adjustable speed clutches taken from manufacturer's catalogues and from Reference (27). These tables point out a chief disadvantage of the fluid clutch; its extremely low torque-to-weight and torque-to-volume ratios as compared to electromechanical and magnetic clutches.

The traction clutch table gives approximate dimensions and weight and the horsepower ratings at different speeds. Torque-to-weight and torque-to-volume ratios using the higher horsepowers and speed are shown in Table 4. Both ratios do not vary a great deal over the entire horsepower range. Since these clutches were designed for industrial purposes only, considerable weight could be saved by the use of light weight metals; but since the power transmitted is proportional to the mass of fluid, they would still have low torque-to-weight ratios because of the need of a high density fluid. Also, the diameter has the greatest effect on the torque so the volume could not be reduced significantly.

Table 2. Traction Clutch Selection Data

Size: OD (in)	Size: Length (in)	Size: Weight (lb)	Horsepower at Various Speeds (rpm): 3600	1800	1750	1620	1450	1200	1170	1080	970	870	810	720
7	2 1/4	2.7	5	3/4				1/2						
13	9	67			20-30	15-25	15-20		7-10	5-7	5	3	2-3	
14 1/2	9	83			40-50	30-40	25-30		15	10-15	7-10	5-7	5	
16 1/2	10	102			60-75	50-60	40-50		20-30	20-25	15	10	7-10	
18	11	125			100		60-75		40-50	30-40	20-25	15-20	15	
19 1/2	11	150			150		100		60-75	50-60	30-40	25-30	20-25	
21	12 1/2	180					150		100	75	50-60	40-50	30-40	
22 1/2	12 1/2	210							125-150		75	60	50-60	30-40
26	15	290							200		100-150	75-125	75	50-75
28 1/2	19	420									200	150-200		100-125

Table 3. Adjustable Speed Clutch Selection Data

Power (hp)	Speed (rpm)	Length (in)	Width (in)	Height (in)	Slip (%)	Control Force (lb)	Load** (variable, constant, or both)
1/3	1150	-	-	-	-	-	Constant
3/4	1750	-	-	-	-	-	Constant
1*	1800	16	15	20	7	10	Both
1	1150	12	15	10	-	-	Constant
1 1/2*	1800	16	15	20	2	10	Both
2*	1800	16	15	20	2.7	10	Both
2	1750	12	15	19	-	-	Constant
3*	3600	16	15	20	3.7	10	Both
5*	3600	16	15	20	4.9	10	Both
7 1/2*	3600	16	15	20	6.6	10	Both
7 1/2	2000	26	28	30	2.6	18	Both
10*	3600	16	15	20	1.7	10	Both
10	1200	26	28	30	3.2	18	Both
15*	3600	16	15	20	2.5	10	Both
15	3500	12	15	19	-	-	Constant
15	1200	26	28	30	4.3	18	Both
20*	3600	16	15	20	3.4	10	Both
20	1200	26	28	30	5.7	18	Variable
20	1200	26	28	30	2.4	32	Constant
25*	3600	16	15	20	4.2	10	Both
25	1200	26	28	30	2.8	32	Both
30	3500	12	15	19	-	-	Constant
30	1800	26	28	30	2.9	25	Both
40	3500	12	15	19	-	-	Constant
40	1800	26	28	30	3.6	25	Both

*Cooler may not be needed.
**Definitions are given on page 47 .

Table 3. (Continued)

Power (hp)	Speed (rpm)	Length (in)	Width (in)	Height (in)	Slip (%)	Control Force (lb)	Load** (variable, constant, or both)
50	1800	26	28	30	4.3	25	Both
60	1800	26	28	30	2.2	44	Constant
60	1800	26	28	30	5.1	25	Variable
75	3500	12	17	23	–	–	Constant
75	1800	26	28	30	2.6	44	Both
100	3500	15	17	23	–	–	Constant
100	1800	26	28	30	3.2	44	Both
125	1800	32	32	39	2.0	61	Constant
125	1800	26	28	30	4.0	44	Variable
150	1800	32	32	39	2.2	61	Both
200	3500	14	20	24	–	–	Constant
200	1800	32	32	39	2.9	61	Both
200	1750	19	26	32	–	–	Constant
300	1800	32	32	39	4.4	61	Both
400	1800	40	40	45	2.5	140	Both
500	3500	19	26	32	–	–	Constant
500	1800	40	40	45	3.1	140	Variable
500	1750	22	30	40	–	–	Constant
500	1200	44	46	53	2.4	350	Constant
600	1200	44	46	53	3	350	Both
700	1200	44	46	53	3.5	350	Both
800	1800	50 1/2	45	49	2.5	110	–
800	1200	44	46	53	4.1	350	Variable
800	1200	53	45	49	3.75	90	–
800	720	75	66	71	1.75	90	–
900	1800	50 1/2	45	49	2.75	110	–
900	600	75	66	71	3.5	80	–
1000	900	64	57	62	2.5	95	–
1000	600	82	76	83	2	85	–

Table 4. Torque-to-Weight and Torque-to-Volume Ratios for Traction Clutch

Horsepower	Speed (rpm)	Torque per Unit Weight, (in-lb/lb)	Torque per Unit Volume, (in-lb/in^3)
5	3600	32.4	1.07
30	1750	16.1	0.89
50	1750	21.7	1.19
75	1750	26.4	1.30
100	1750	28.8	1.33
150	1750	36.0	1.68
150	1450	36.2	1.49
150	1170	38.5	1.58
200	1170	37.1	1.34
200	970	31.0	1.35

An approximate 12 to 15 per cent reduction in the outside diameter of the rotating parts can be realized by using the twin-type design, but the length is increased. In this design two clutches are enclosed in the same housing. The axial thrust of one working circuit opposes the other and they balance out.

Power at rated slip, approximate dimensions, scoop tube control force, and the type of load are given in the adjustable speed table. Variable loads are those whose horsepower varies as the cube of the speed, whereas, in constant loads the horsepower varies directly with the speed.

Adjustable speed clutches generally have lower torque-to-weight and torque-to-volume ratios than the traction type because of the necessity of the outer housing, reservoir, and cooler.

The pounds force required to move the scoop tube shows a significant increase as the power rating increases. Lever, handwheel, or a sliding mechanism control are available for scoop tube positioning. By the application of automatic control to the scoop tube, the output speed can be maintained at a constant speed as close as plus or minus 1 per cent.

Although a clutch can be designed to operate at a slip of 1 per cent, generally it is not economical to design for less than 1 1/2 per cent. Referring to Table 3, the slip ranges from around 2 per cent to about 6 per cent at rated horsepower. Since the power lost to slip goes into heat, it is desirable to keep the slip as small as possible.

Fluid Qualities

Since the transmitted torque is through the liquid in the clutch, the fluid used should possess certain qualities. It should have low viscosity to reduce friction, but the viscosity must be high enough to provide lubrication for the bearings. A high density is desirable since the torque capacity of the clutch depends on the flow rate.

As the traction clutch may run in a stalled condition for short periods, a relatively high flash point is required to reduce the fire hazard if flammable fluids are used. Resistance to oxidation and formation of sludge deposits when subjected to the affects of time, temperature, and turbulence is also desirable. To prevent the mixing of air with the fluid, it is recommended that the fluid be foam suppressing. Other required qualities are that the fluid be non-corrosive and have a low freezing point.

Although water and mercury have been used in the fluid clutch, a study of all factors involved shows that petroleum oils having a viscosity of around 150 Saybolt seconds at 100°F are best suited for industrial applications because of their lubricating qualities, availability, and the high overall efficiency obtained.

Example Using Cryogenic for Clutch Fluid

Cryogenic fluids lack many of the desired characteristics wanted in a fluid for the clutch; but since they are being used on board space vehicles, the possibility of their use for fluid clutches should be considered.

Materials of construction offer no problems as 300-series stainless steels, titanium, and aluminum are acceptable for cryogenic service (28, 29,30). Work has also been done on the use of cryogenic fluids for lubricating bearings (31).

In this case the prevention of heat transfer to the clutch is of prime importance to reduce the evaporation of the liquid cryogenic.

Careful design of materials to minimize the contact area of metals can reduce the transfer by conduction. A high vacuum environment and multiple blanket type insulation, such as SI-62 which consists of alternate layers of 2.5×10^{-4} inch aluminum foil on thin fiberglass paper (32), can minimize transfer by radiation.

The power lost to slip between the primary and secondary elements is converted into heat. This eliminates the traction type clutch from consideration as a continuous supply of cryogenic fluid must be pumped into the working circuit and means provided to remove the vapors resulting from the slip loss.

There are presently five cryogenic fluids of importance for space applications. These are oxygen, fluorine, nitrogen, helium, and hydrogen. Since aluminum and titanium are explosive under certain conditions in an atmosphere of oxygen or fluorine, oxygen and fluorine will not be considered desirable as clutch fluids. Both helium and hydrogen suffer from very low densities. Nitrogen will be used in an example since it is relatively inert, nonflammable, nontoxic, and a viscosity graph was readily available.

Assume a fluid temperature of 70°K. At this temperature nitrogen has a viscosity of 0.0022 poises and a density of 52.4 lbs/ft^3 (33). If the friction factor equations, (3.4d) and (3.5), represent the case with cryogenic fluids, then the torque transmitted through the clutch with liquid nitrogen can be determined using the method in Chapter III.

The results for the test clutch are shown in Figure 17 along with a torque-slip curve for the oil whose viscosity was 190 Saybolt seconds (0.185 poises) at 100°F. The low viscosity of the cryogenic leads to a

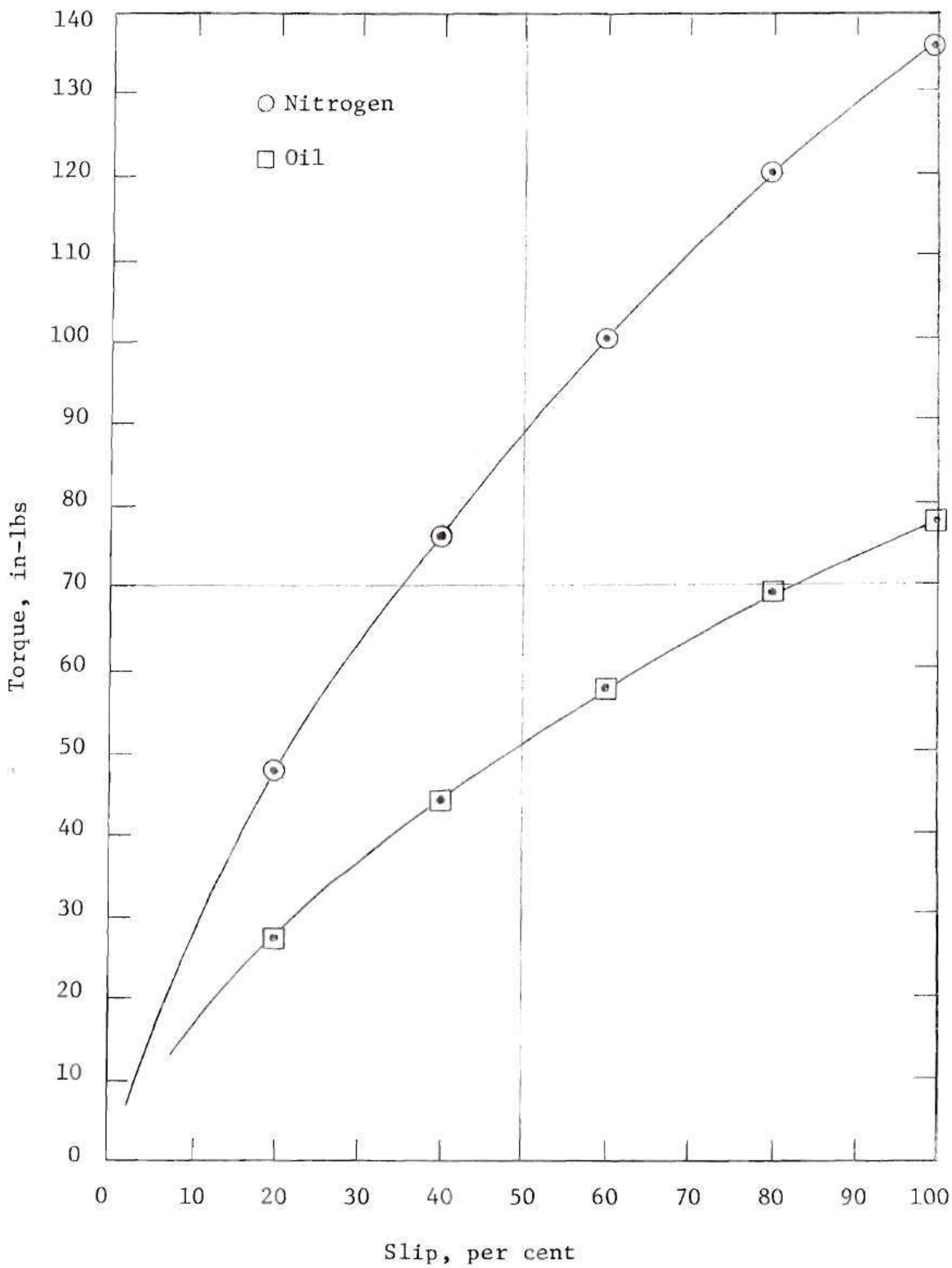


Figure 17. Comparative Torque-Slip Curves for Cryogenic and Oil.

considerable torque increase over the higher viscous oil. Input speed was 1800 rpm.

CHAPTER VI

CONCLUSIONS

Fluid clutches offer mechanical simplicity, light-load starting, limited stalling torque, and considerable flexibility in alignment. In addition, the adjustable speed type offers complete declutching of the load from the driver and acceleration control from seconds to minutes with infinitely variable speed regulation. Properly designed clutches can also reduce input torque fluctuations by 98 per cent, but improper designs can greatly increase these fluctuations. At one point the test clutch increased input fluctuations eleven times.

Fluid clutches suffer from slip which results in the necessity of continuous heat dissipation from the clutch. This heat is dissipated by convection from the traction clutch. Adjustable speed clutches are generally furnished with heat exchangers. For vacuum conditions the mode of heat transfer from the traction clutch is radiation. Both radiation and conduction should be considered for the adjustable speed clutch.

Torque-to-weight and torque-to-volume ratios are low. For the traction clutch these ratios remain fairly constant over the entire horsepower range. The torque-to-weight ratios range from 16.1 to 38.5 in-lbs/lb with an average value of 30.4, and the torque-to-volume ratios range from 0.89 to 1.68 in-lbs/in^3 and averaged 1.3. These ratios are somewhat smaller for the adjustable speed clutch because of the necessity of an outer housing, reservoir, and cooler.

The large diameters of the impeller and turbine and the high density fluids needed for increased torque capacity result in high inertia.

Excessive turbulence and losses at high slip values produced unpredictable torque outputs in the test clutch. Torque increased with slip until reaching approximately 50 per cent slip after which the torque followed a path of unpredictable nature at a torque less than the maximum obtained at the slip value of around 50 per cent.

Temperature increases which reduce the viscosity of the fluid are beneficial as long as the safe operating temperature of the fluid is not exceeded. In one test a temperature increase of 75 per cent resulted in a 25 per cent torque increase.

Transmitted torque is dependent on the quantity of fluid in the working circuit. Tests showed that a change in the amount of fluid in the clutch from 8 oz to 11.2 oz resulted in a doubling of the torque output.

Highly viscous fluids can produce significant shear torque, especially at temperatures below 100°F. This torque was significant up to test clutch speeds of 1400 rpm for an oil whose viscosity was 2390 SUS at 100°F.

Fluid clutches do not appear to have great application in space usage. They could possibly be useful where cryogenic fluids are available. Calculations were carried out for a cryogenic fluid. They showed that the use of cryogenic fluids can result in substantial torque transmission. In a theoretical case for the test clutch, liquid nitrogen produced 1 3/4 times more torque than an oil. However, the evaporation of the cryogenic would require a continuous supply of fluid.

APPENDIX

APPENDIX A

INSTRUMENTATION AND EQUIPMENT

A schematic of the test apparatus is shown in Figure 18. A detailed description follows.

Hydraulic Drive Apparatus

Fluid at 750 psi was supplied to the Moog servo valve, model 22-135A, by a Denison "HydrOILic" pumping unit as shown in Figure 19.

The Moog servo valve was used to control the flow to a Vickers hydraulic motor, Model AR-10007-BEL, which furnished power to the system. Since the capacity of the servo valve was insufficient at high speeds, by-pass needle valves were used to bring the system to the approximate desired speed. Final speed adjustment was then made using the servo valve.

To reduce fluctuations in speed a flywheel was used after the hydraulic motor.

Speed Control Apparatus

Speed control was obtained by using a velocity control feedback system (see Figure 20). The reference signal for the feedback system was supplied by a Harrison Laboratories d.c. power supply, model 6204A.

The filtered output of a Fairchild Industries model 532-A1 d.c. generator was the negative feedback signal. With a 60 tooth gear on the generator shaft and a 112 tooth gear on the shaft following the flywheel, the generator constant was 0.00357 volts output per rpm of the hydraulic

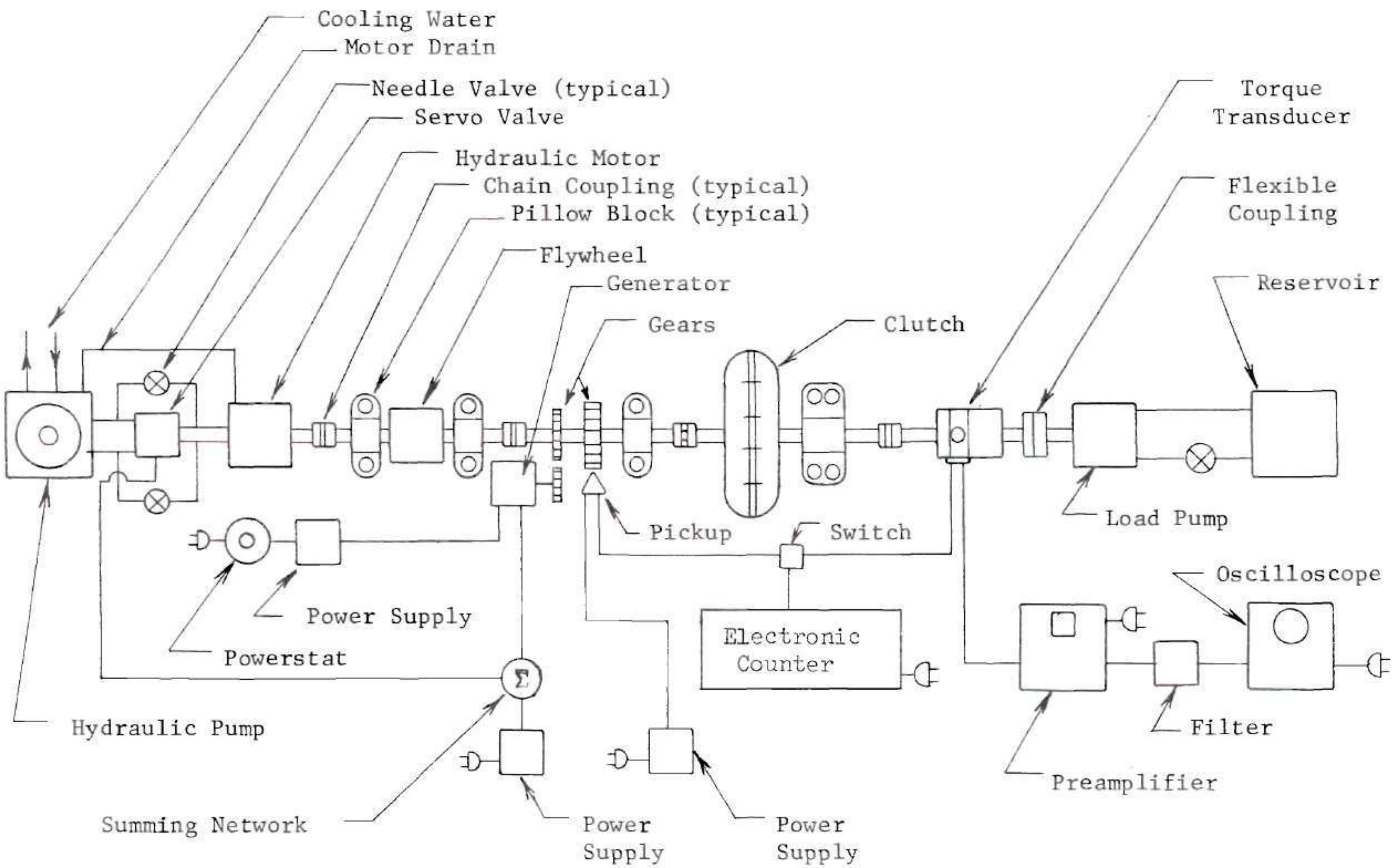


Figure 18. Schematic of the Experimental Test Apparatus.

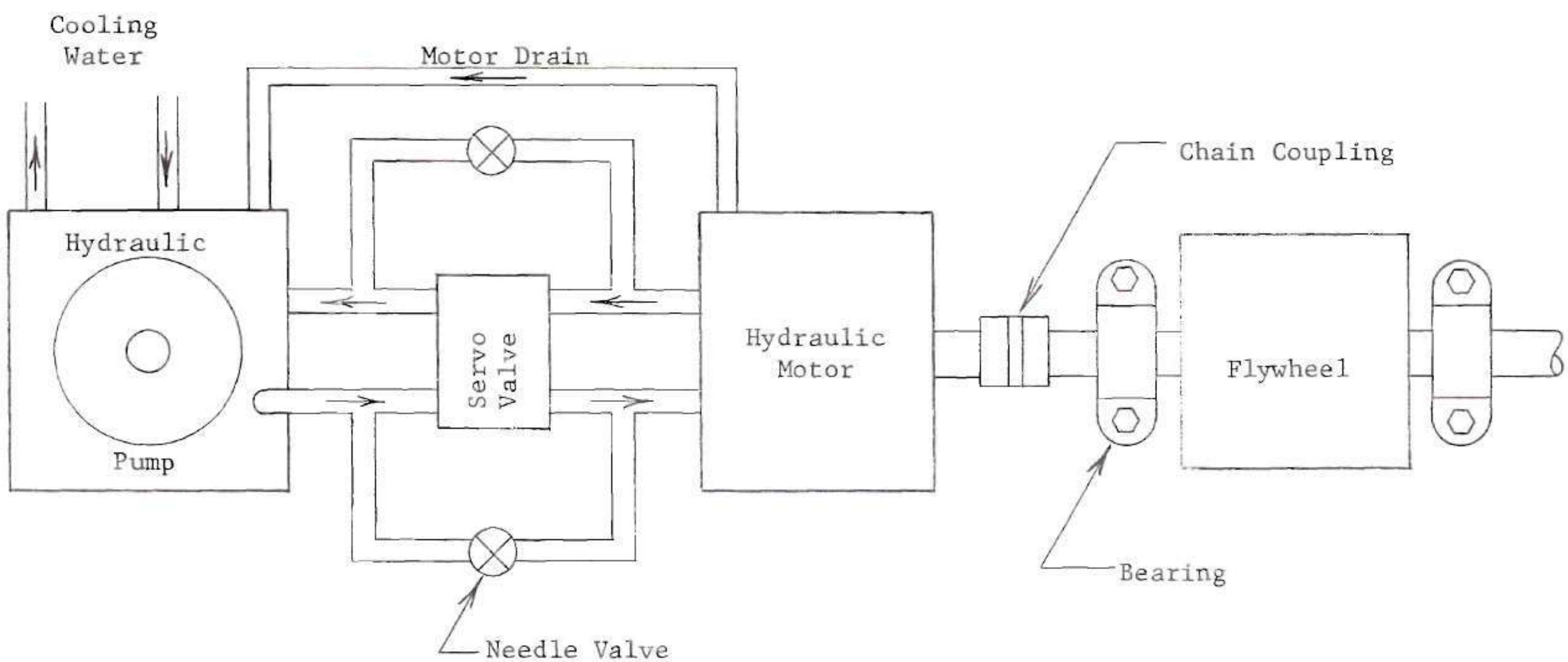


Figure 19. Diagram of the Hydraulic Drive Apparatus.

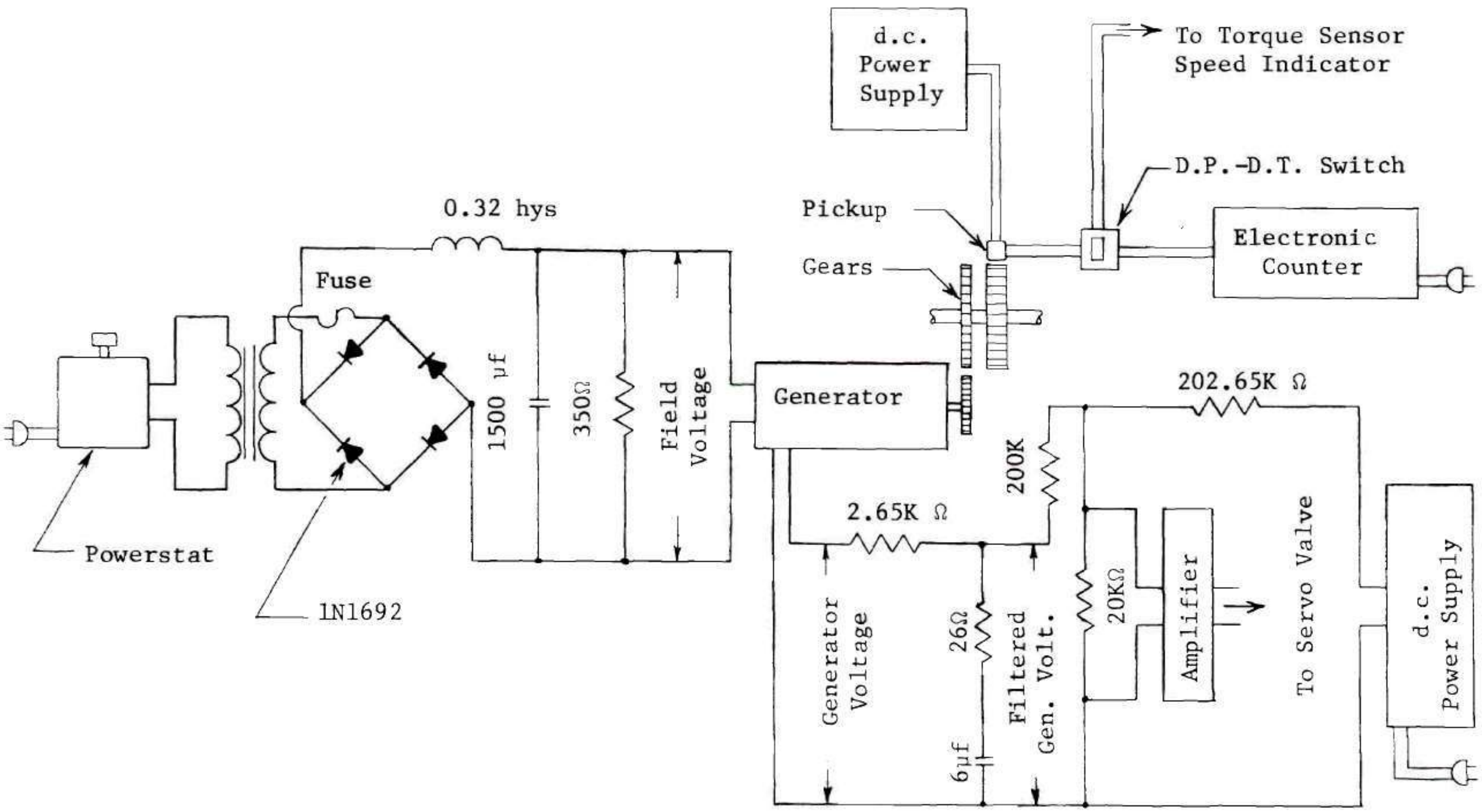


Figure 20. Diagram of the Speed Control Apparatus.

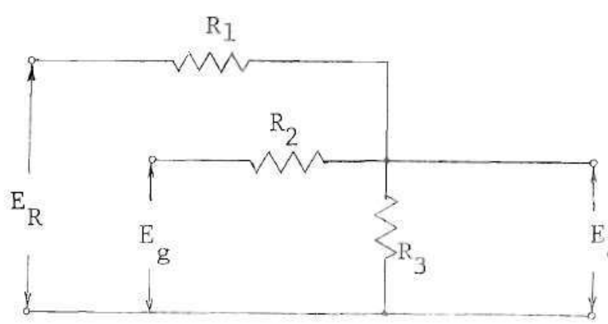
motor shaft when 12 volts were applied to the field. To supply the d.c. field voltage to the generator, a silicon diode power supply was made as shown on the left of Figure 20.

A filter network follows the generator to filter the varying output voltage. This filter's transfer function is

$$\frac{E_{out}}{E_{in}} = \frac{\tau_1 s+1}{\tau_2 s+1}$$

where $\tau_1 = 0.000156$ and $\tau_2 = 0.016$.

The feedback signal from the generator was summed with the reference voltage using a circuit as shown in Figure 21.



$$E_o = \frac{R_3(R_2E_R + R_1E_g)}{R_1R_2 + R_1R_3 + R_2R_3}$$

Figure 21. Summing Circuit.

E_R is the reference voltage and E_g is the filtered generator voltage. Using the values of resistors from Figure 20, the summing circuit output, E_o, was approximately

$$E_o \approx \frac{(E_R + E_g)}{12}$$

Note that E_g was a negative voltage. The actuating signal from the summer was amplified by a Dymec amplifier (model 2460A), and the output feed to the servo valve to control the fluid flow to the motor.

A Hewlett-Packard 52331 electronic counter was used to display both the input speed to the clutch and the output speed. Input speed was obtained by a model 836 photosensitive pickup, supplied by Power Instruments, focused on a 120 tooth spur gear on the flywheel shaft. The top land of the gear teeth were polished while the face, flank, and bottom land were blued. The pickup gave a pulse each time a gear tooth passed in front of it, and the pulses were summed and displayed by the Hewlett-Packard counter. This displayed number had to be divided by two to determine the rpm since the pulses were displayed each second and the gear had 120 teeth.

Power for the photosensitive pickup came from one-half of a Kepo (model 430D) voltage-regulated dual d.c. power supply.

The built-in speed sensor of the Lebow rotating shaft torque sensor was used for the output speed of the clutch. It consisted of a magnetic pickup focused on a 60 tooth gear mounted on the torque sensor shaft. Output of the magnetic pickup went to the electronic counter. Since the gear used for this pickup had 60 teeth and the counter displayed the number of pulses per second, the number displayed was the rpm of the clutch output shaft.

A double-pole, double-throw switch had to be used to read successively the clutch input and output speed since the Hewlet-Packard counter did not have a dual input.

Torque Measuring Apparatus

The test clutch was positioned after the photosensitive pickup gear, and the apparatus used to measure the torque can be seen in Figure 22.

Following the clutch was a model 1214-100 Lebow strain gage bridge torque transducer with a torque capacity of 100 in-lbs. Its output was feed into a Sanborn recorder preamplifier, model 350-1100B, and the amplified output then went through a filter and to a Textronic Type 564 storage oscilloscope with a Type 3172 dual-trace amplifier.

The torque load was applied by the New York Air Brake Company variable delivery pump following the transducer. By adjustment of the valve on the output line, the flow was restricted and the torque varied accordingly over the desired range.

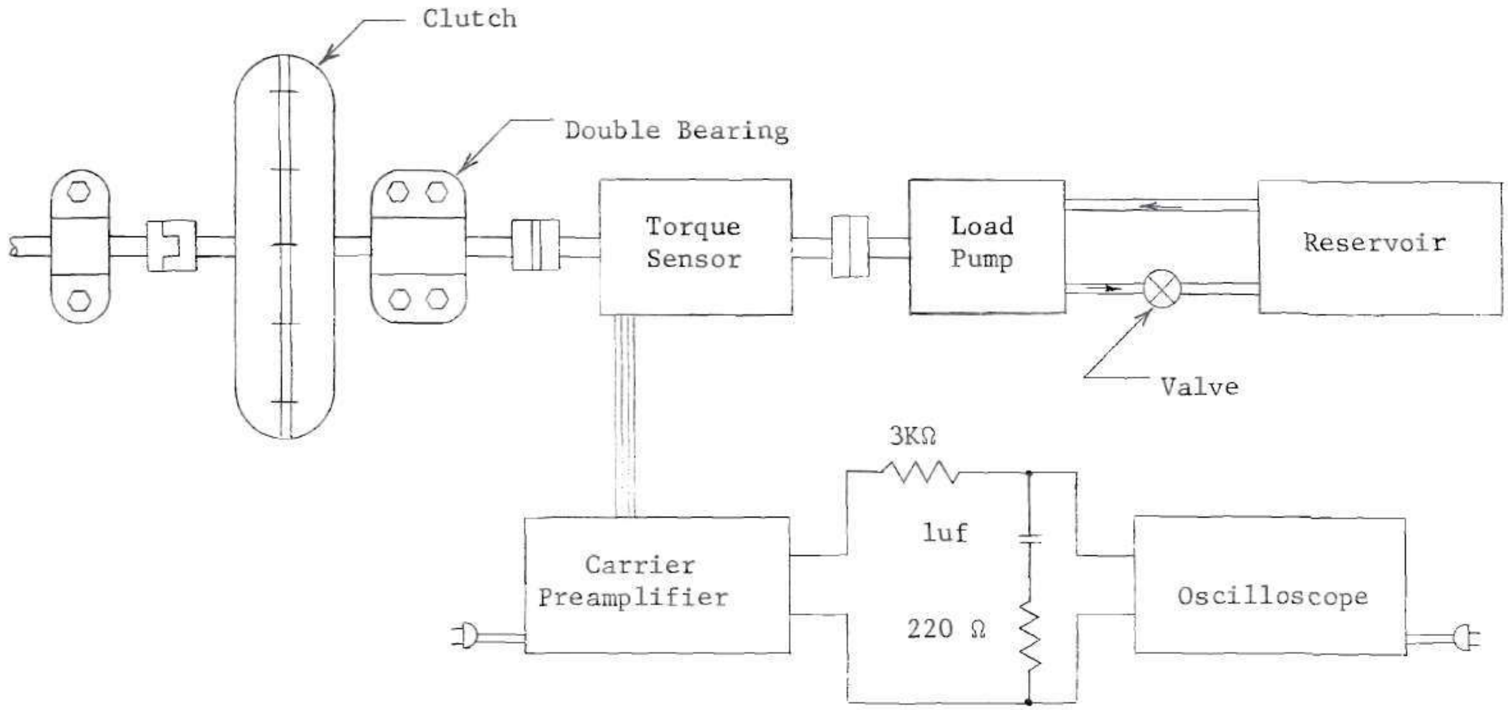


Figure 22. Diagram of the Torque Measuring Apparatus.

APPENDIX B

EXPERIMENTAL DATA

The experimental and theoretical data obtained in this investigation are shown in the curves on the following pages.

The light weight oil (less viscous) used in the tests is designated as "Fluid I" while the more viscous oil is designated as "Fluid II".

Figure 23 shows the effect temperature has on the drag torque. At higher speeds this effect becomes significant.

For a constant temperature, the quantity of fluid in the working circuit determines the torque capacity. Figure 24 shows that the addition of 3.2 ounces of fluid can double the torque at 1800 rpm.

A theoretical torque-slip curve is shown in Figure 25 along with the experimental curve for the same conditions. The theoretical curve was calculated using the method of Chapter III. Temperature was 100°F and input speed was 1800 rpm for both curves.

Figures 26,27, and 28 are torque-slip relations for various degrees of fill of Fluid I. Input speed was varied to show the variation of torque with speed.

Drag torque curves for Fluid II appear in Figures 29 and 30. The 100°F and 125°F curves in Figure 29 show a greater torque over part of the speed range than the other two curves. Large shear torque produced by this high viscous oil at low temperatures accounts for this fact. Figure 30 shows that the torque for 11.2 ounces is only one and one-half times the torque for 8 ounces as compared to two times for Fluid I at 1800 rpm.

Comparative drag torque curves for both fluids are presented in Figures 34 through 39. At temperatures above 100°F the viscosity and shear torque of Fluid II is reduced to the point such that the less viscous oil produces greater torque.

Figures 40, 41, and 42 are torque-slip curves comparing Fluids I and II. Fluid II produces considerable torque over Fluid I for the tests using 5 ounces of oil.

The effects of various clearances between the impeller and turbine are shown in Figures 43-46. As can be seen, the effects are slight except for the data taken using only 5 ounces of oil.

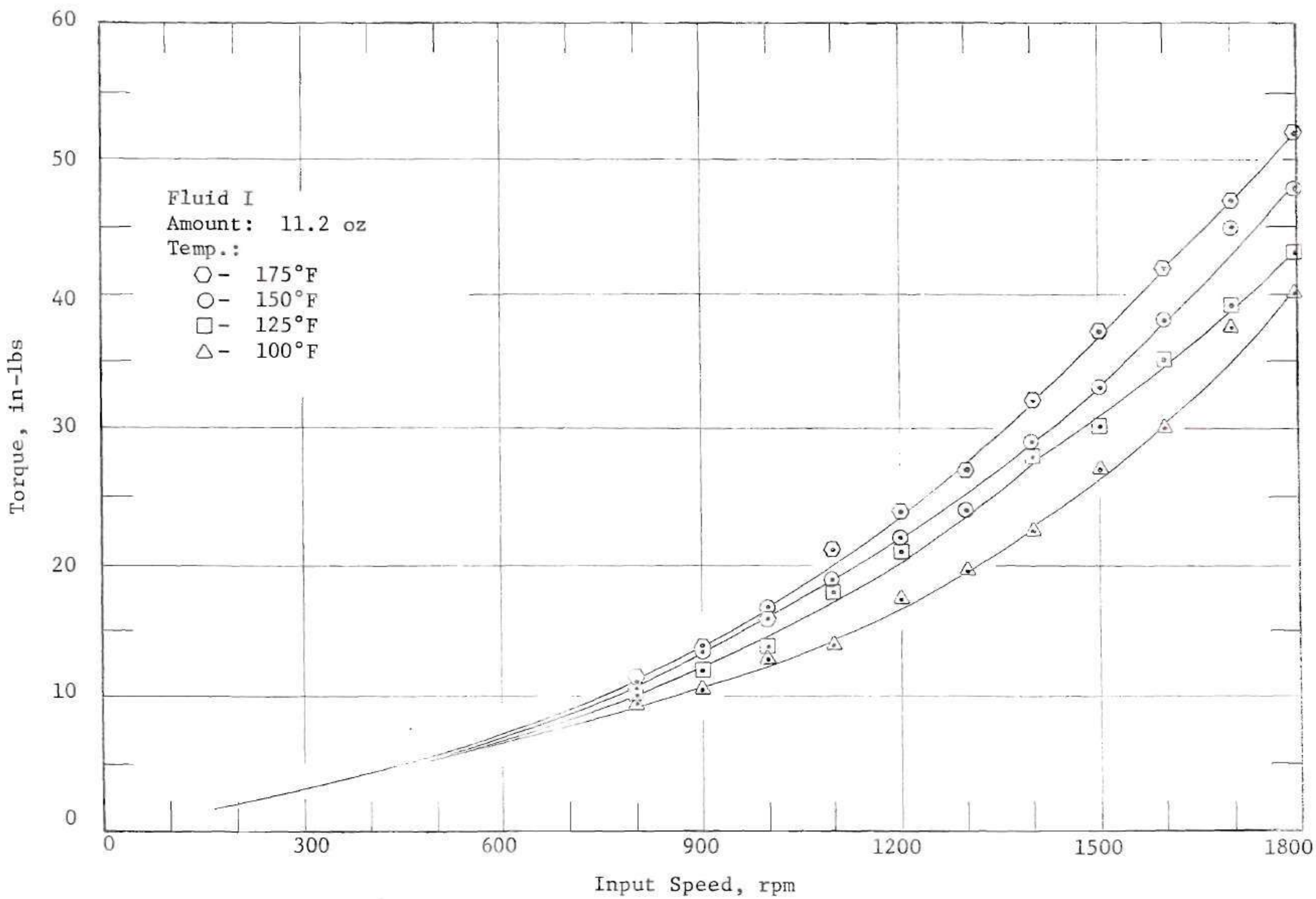


Figure 23. Drag Torque for Fluid I at Different Temperatures.

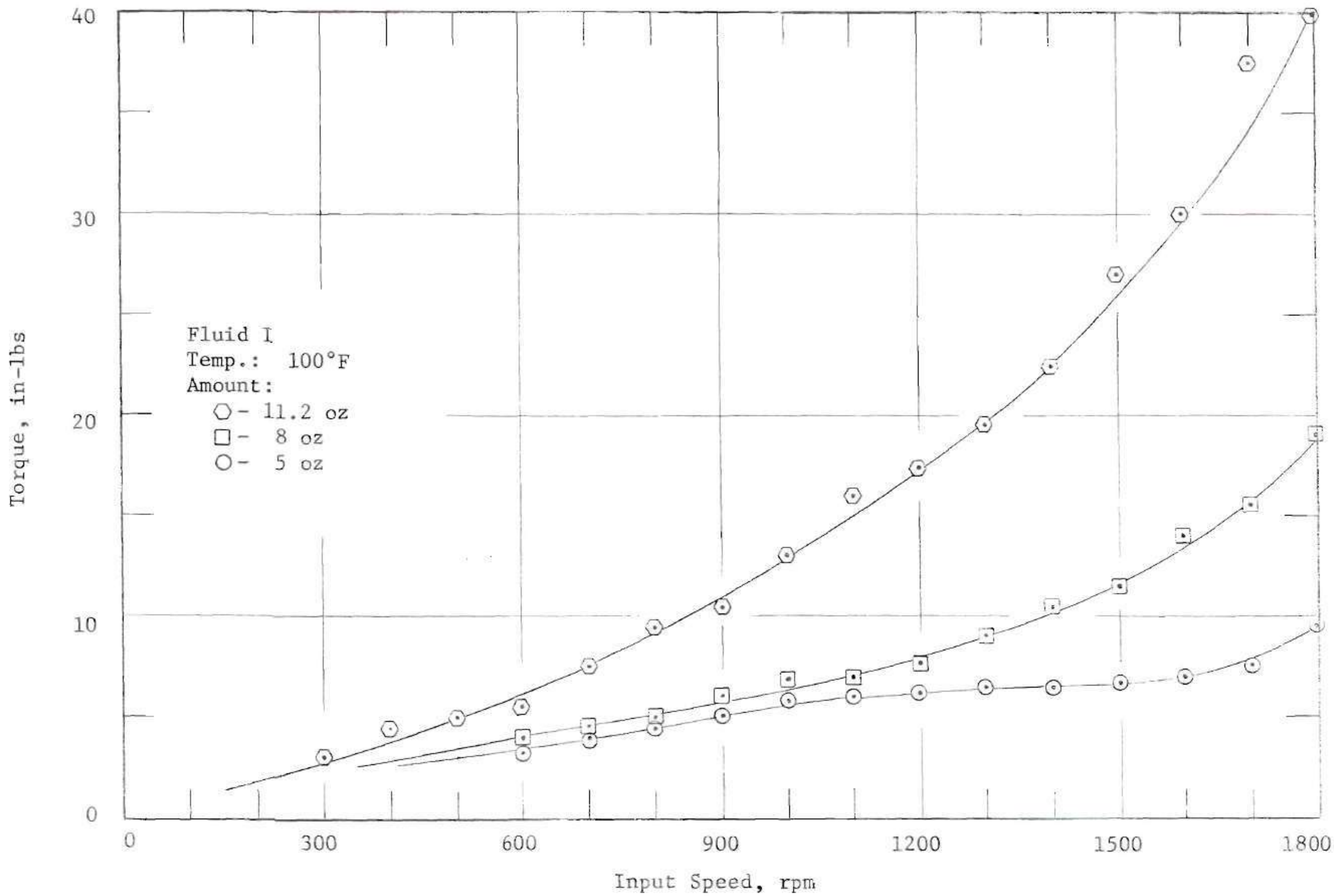


Figure 24. Drag Torque for Fluid I at Various Degrees of Fill.

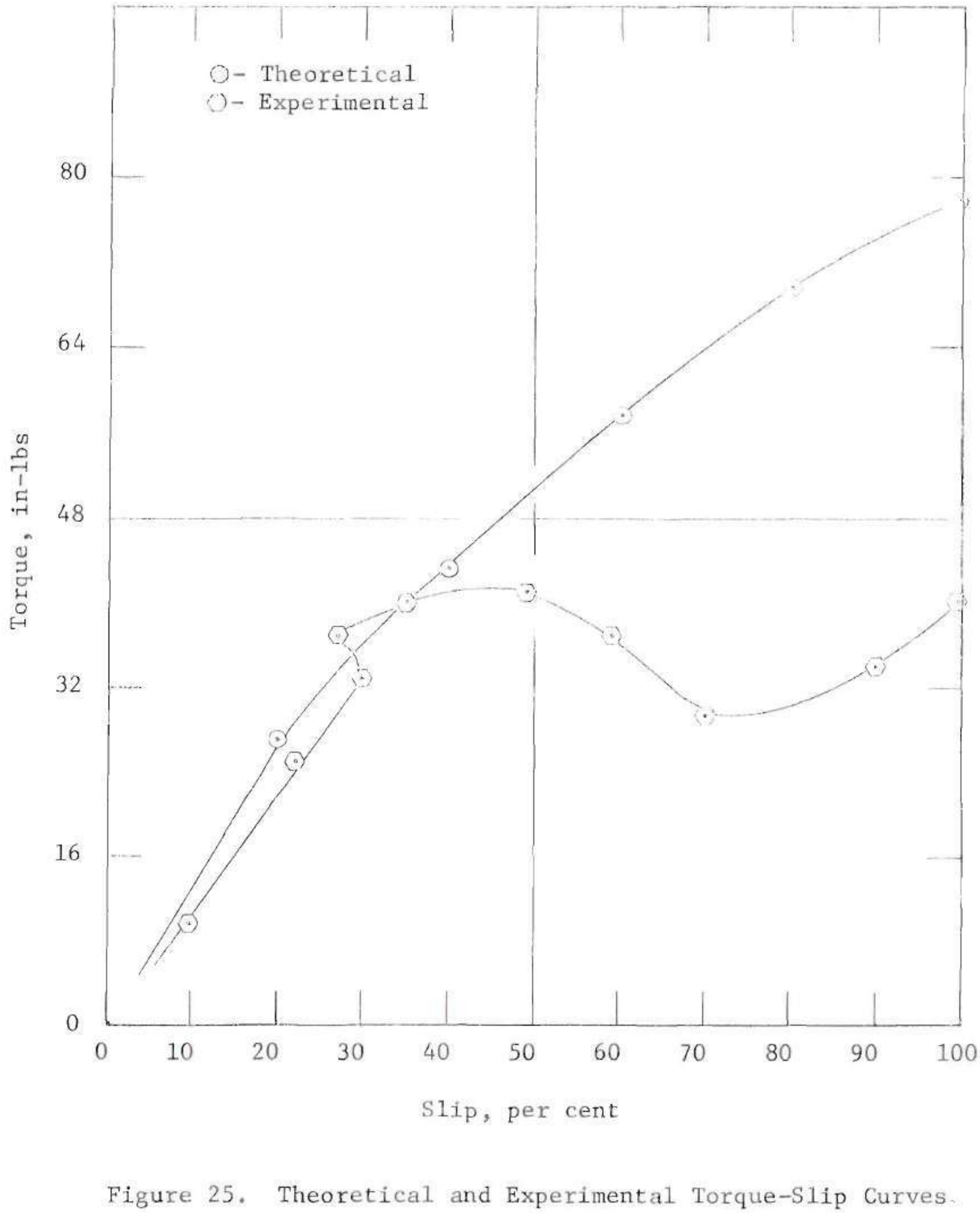


Figure 25. Theoretical and Experimental Torque-Slip Curves.

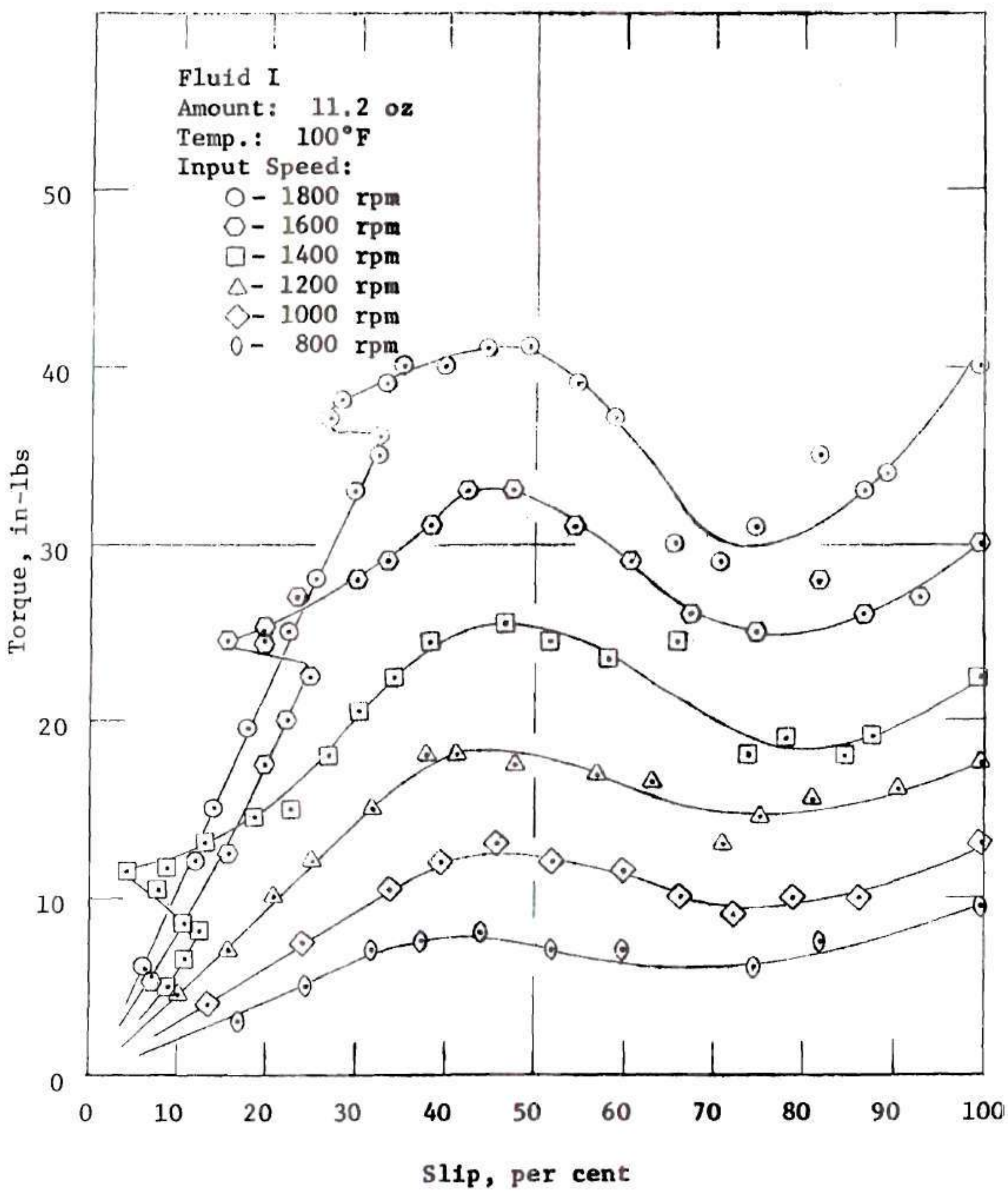


Figure 26. Torque-Slip Curves for Fluid I.

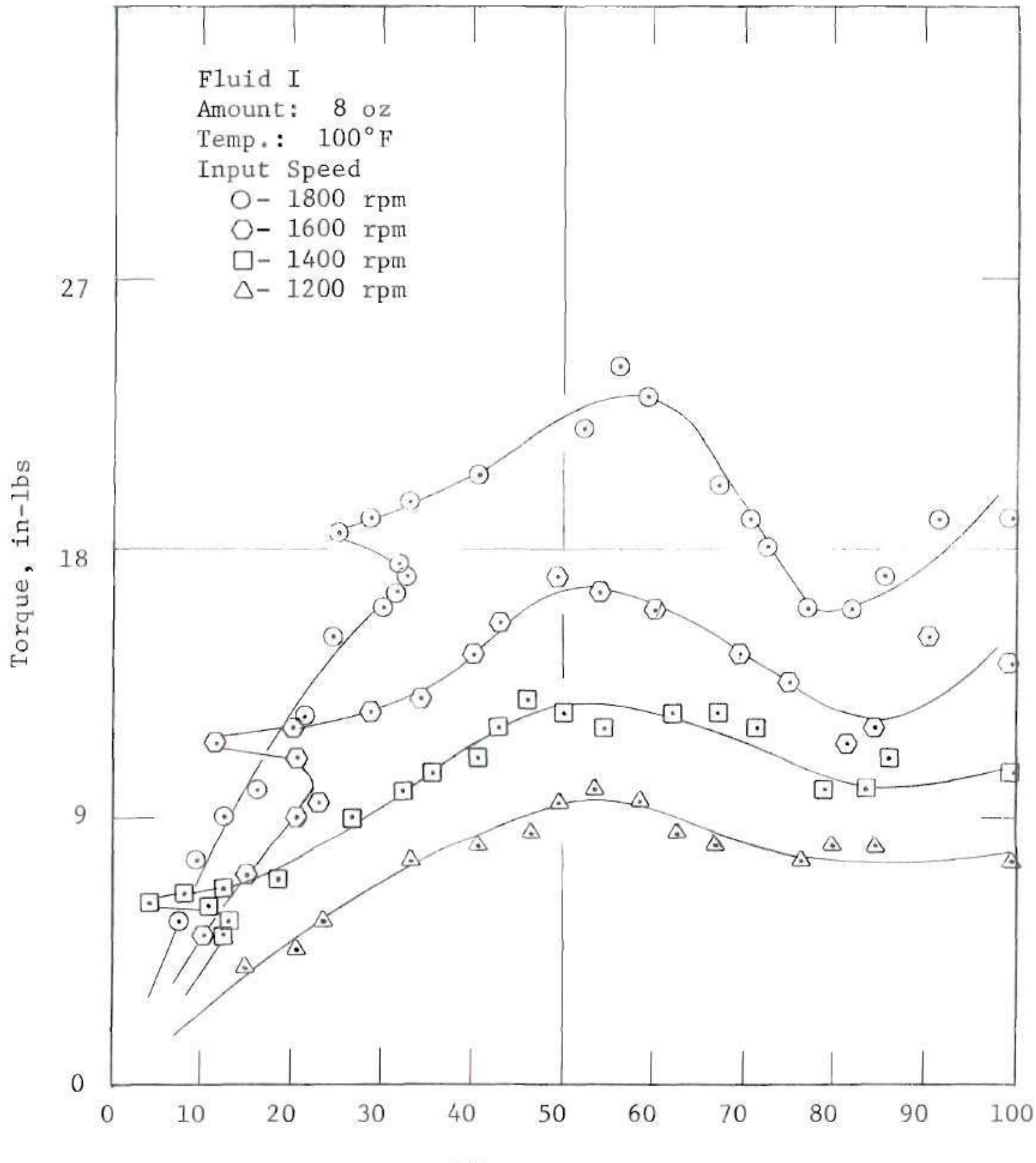


Figure 27. Torque-Slip Curves for Fluid I.

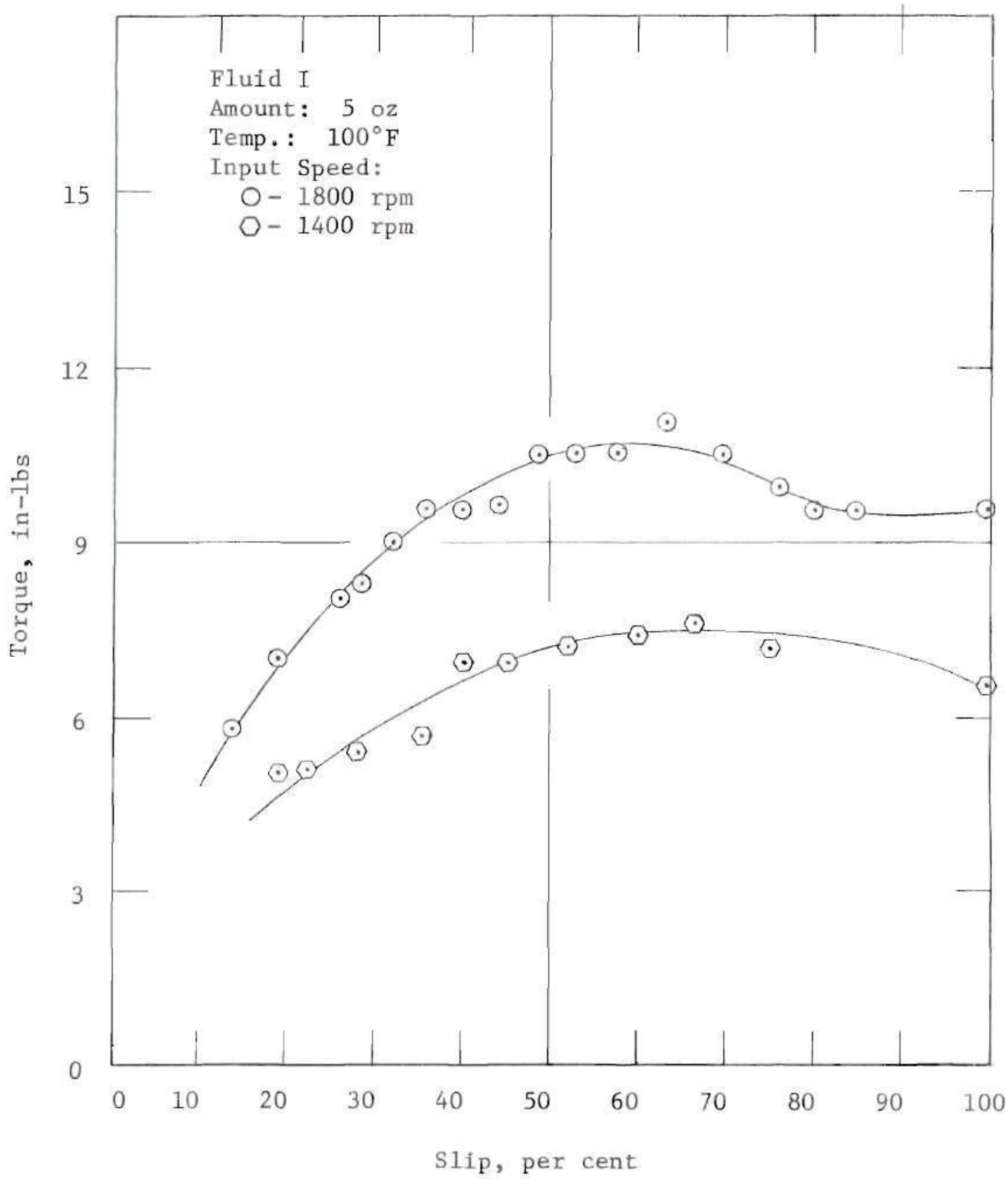


Figure 28. Torque-Slip Curves for Fluid I.

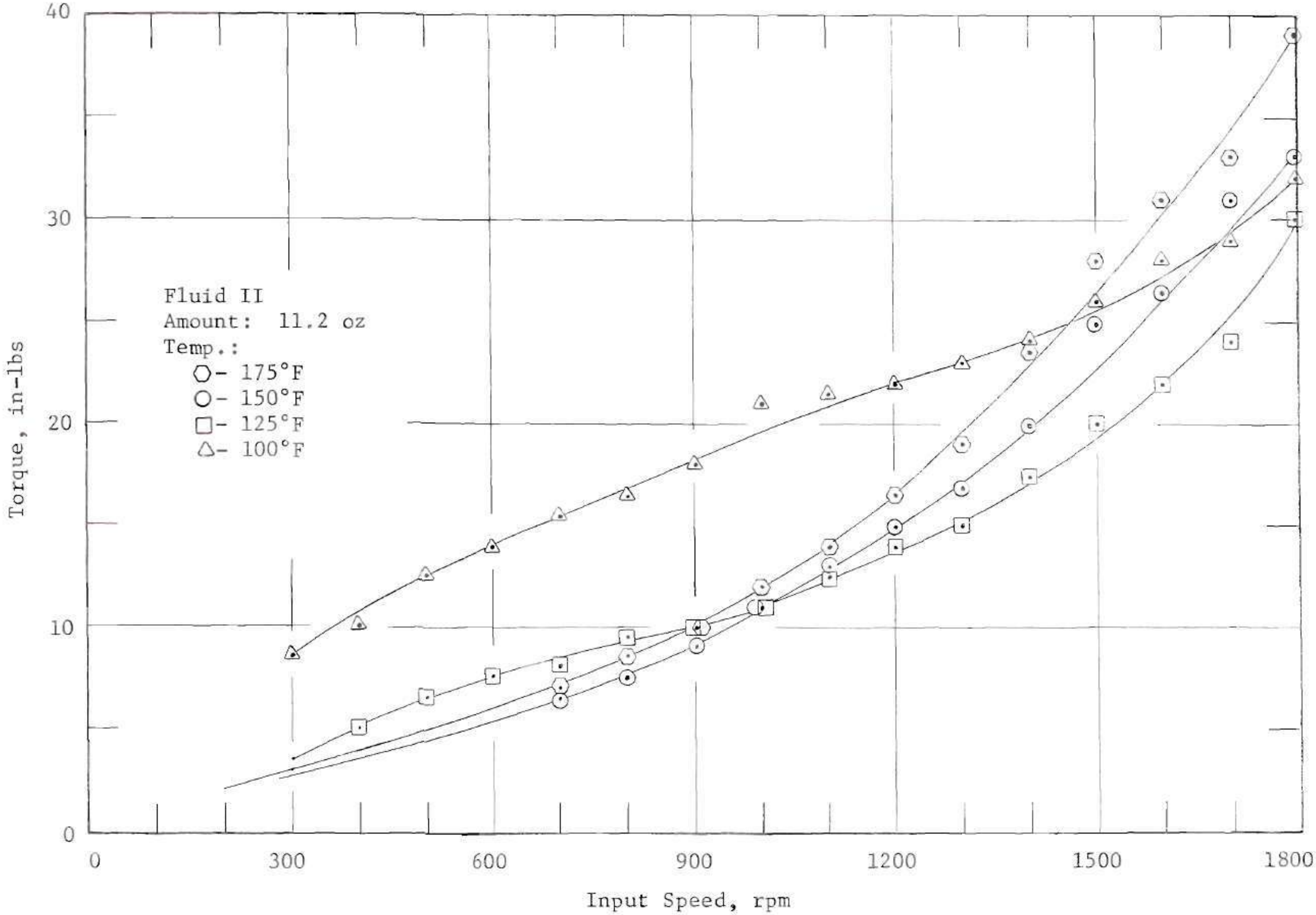


Figure 29. Drag Torque for Fluid II at Different Temperatures.

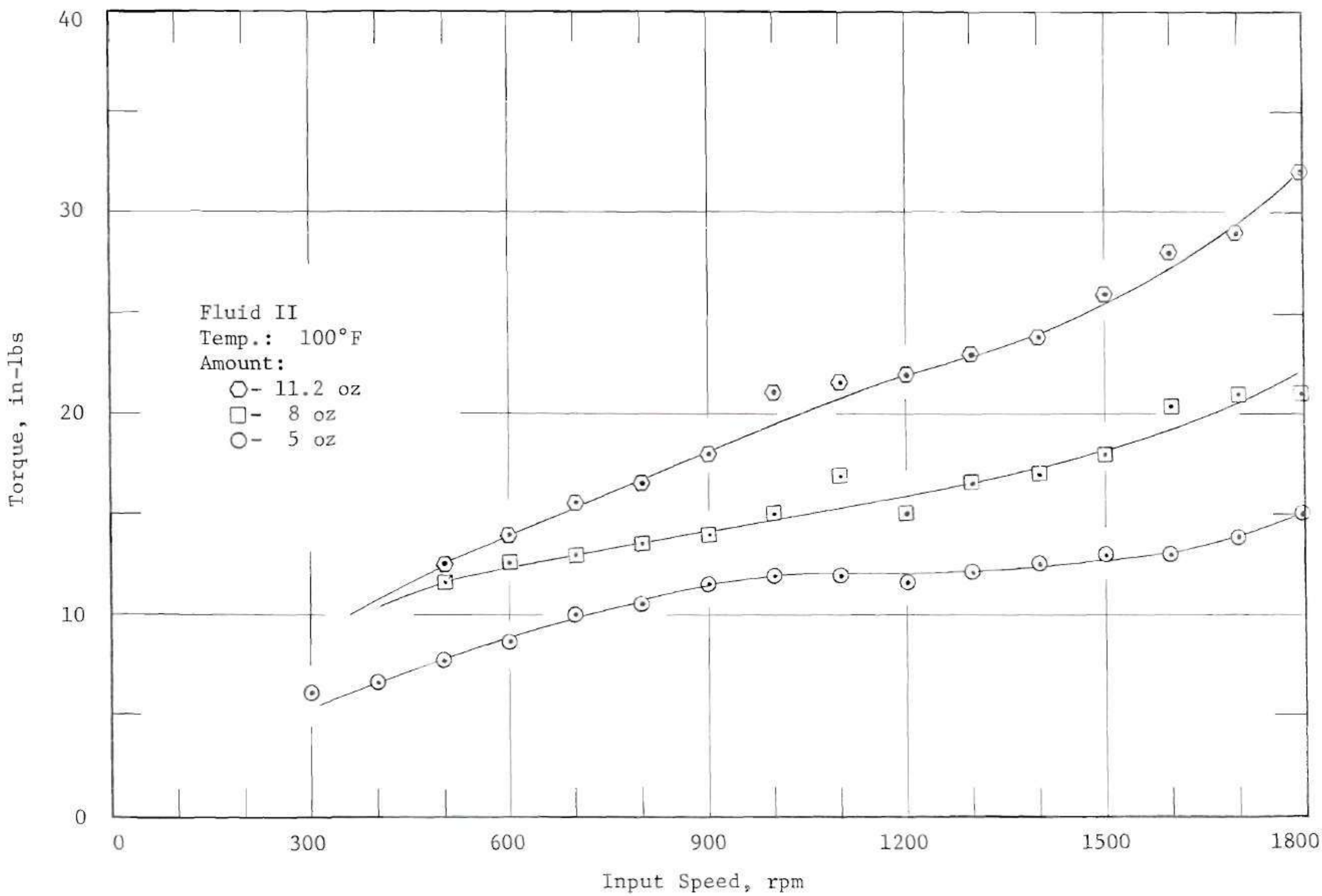


Figure 30. Drag Torque for Fluid II at Various Degrees of Fill.

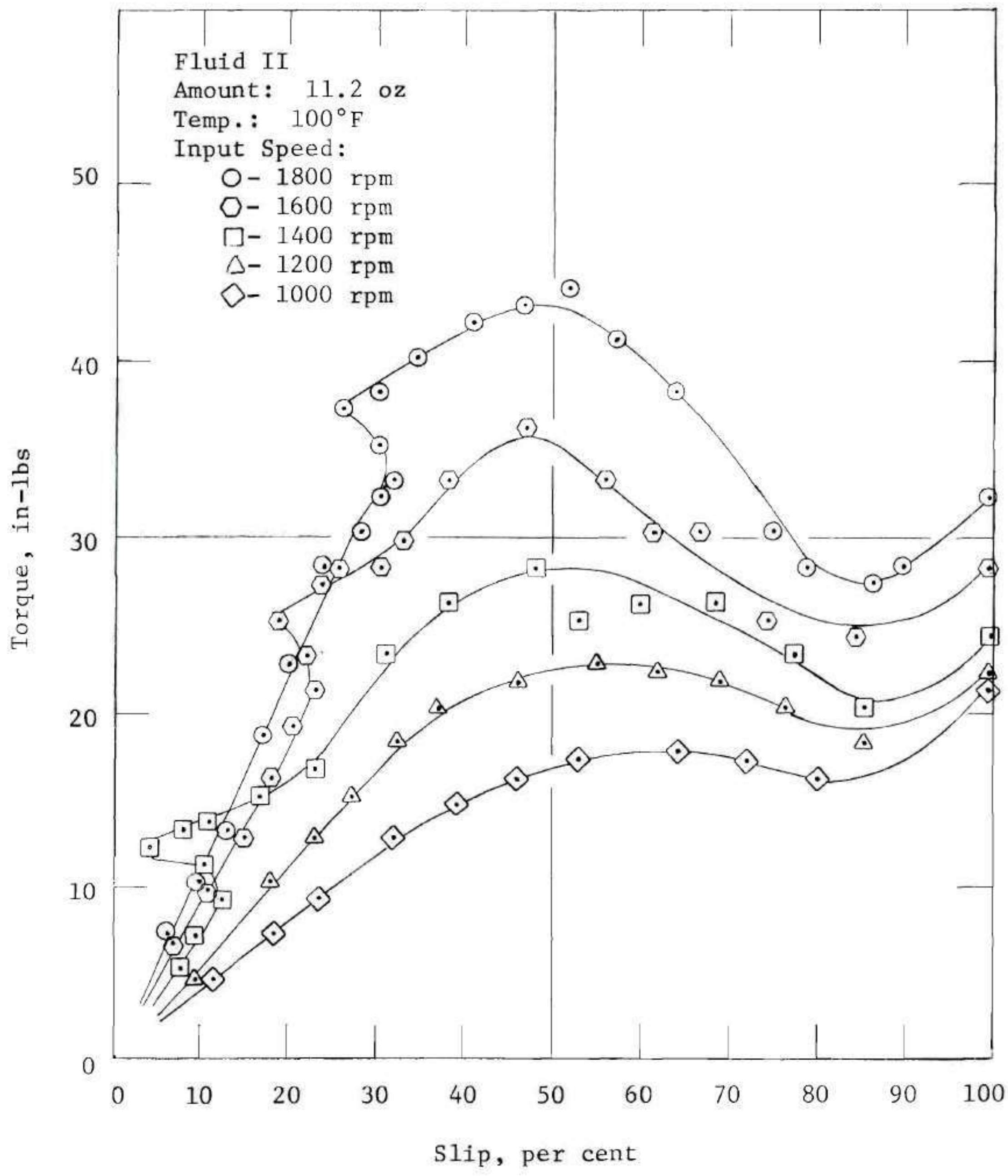


Figure 31. Torque-Slip Curves for Fluid II.

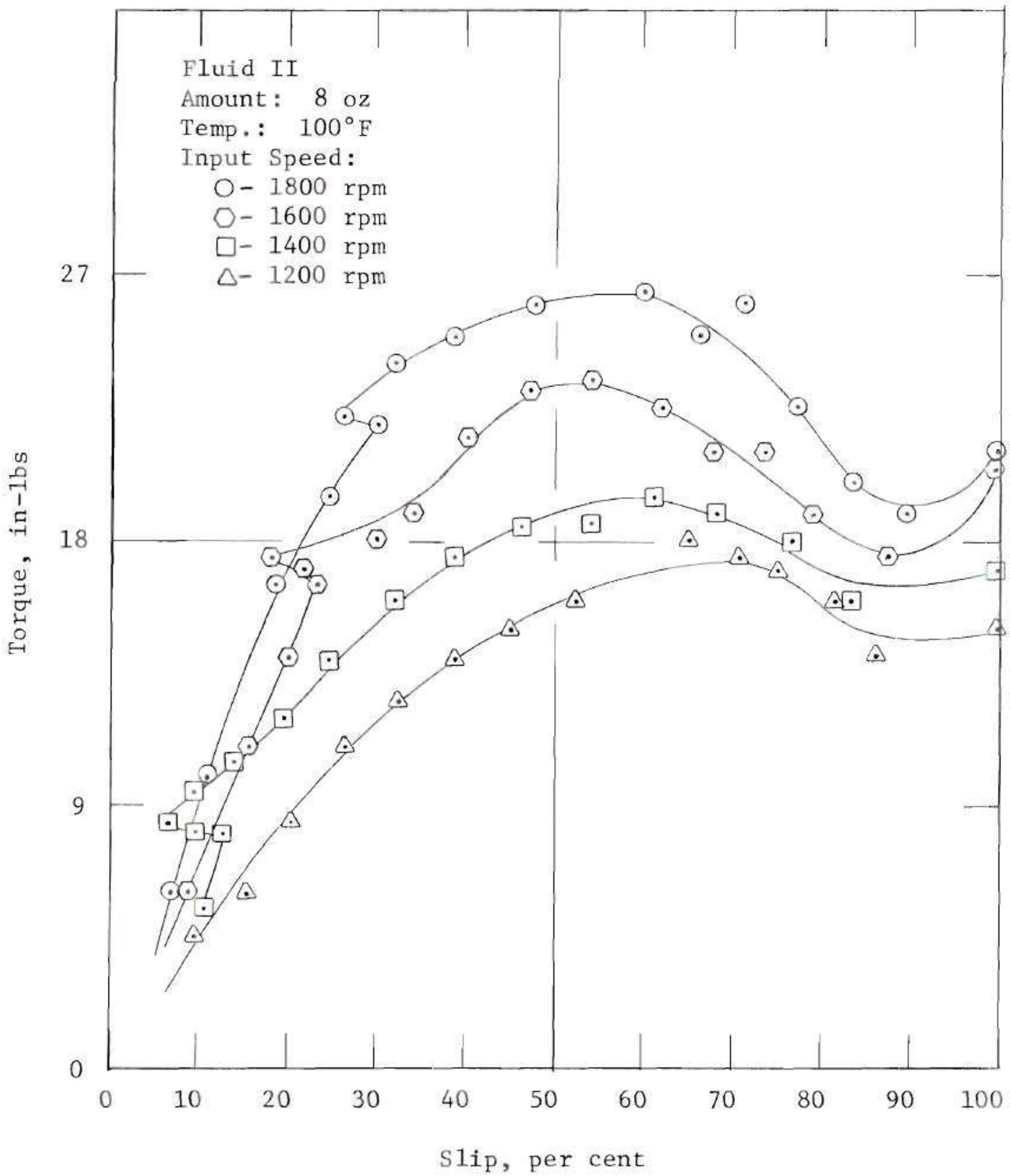


Figure 32. Torque-Slip Curves for Fluid II.

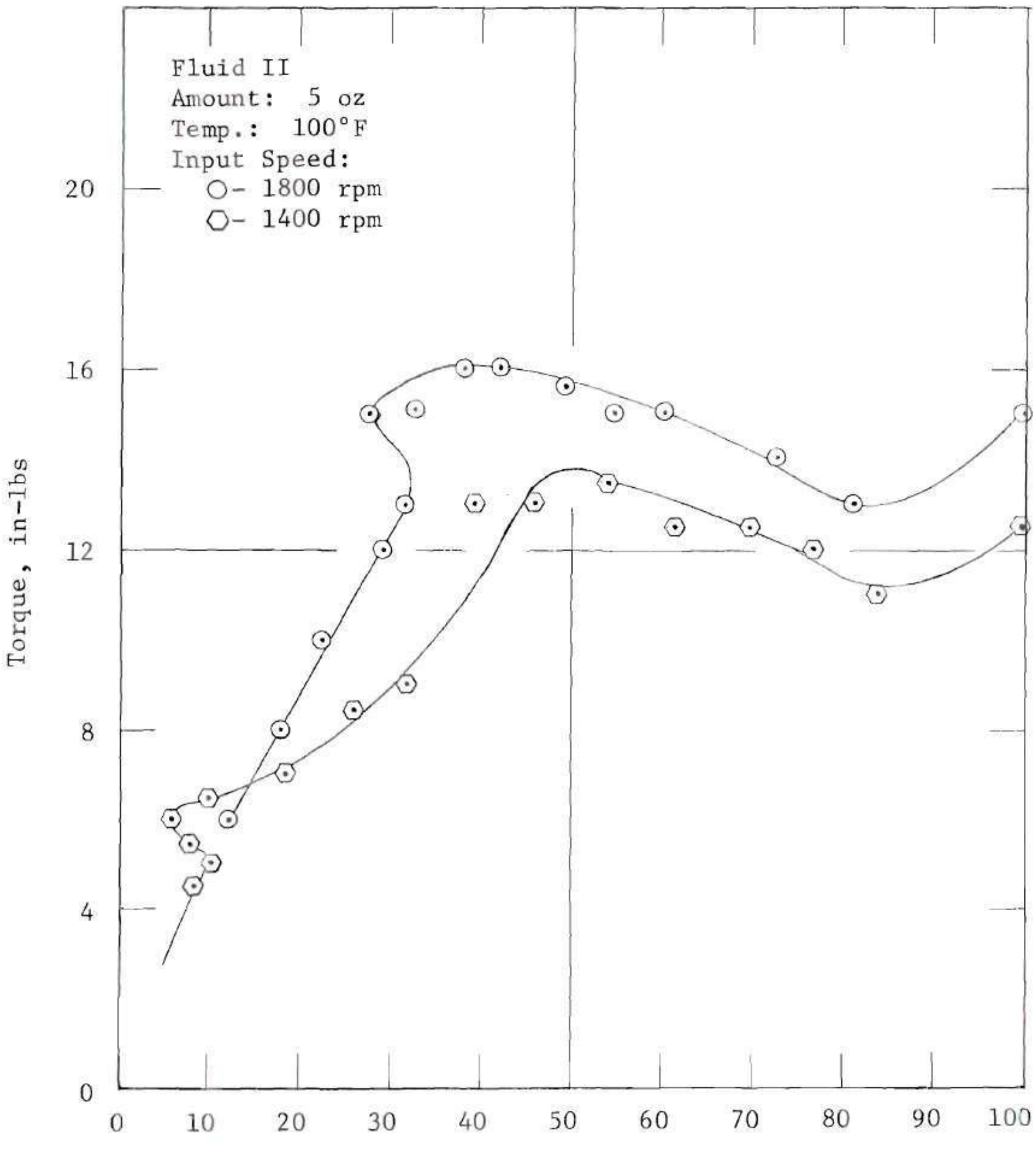


Figure 33. Torque-Slip Curves for Fluid II.

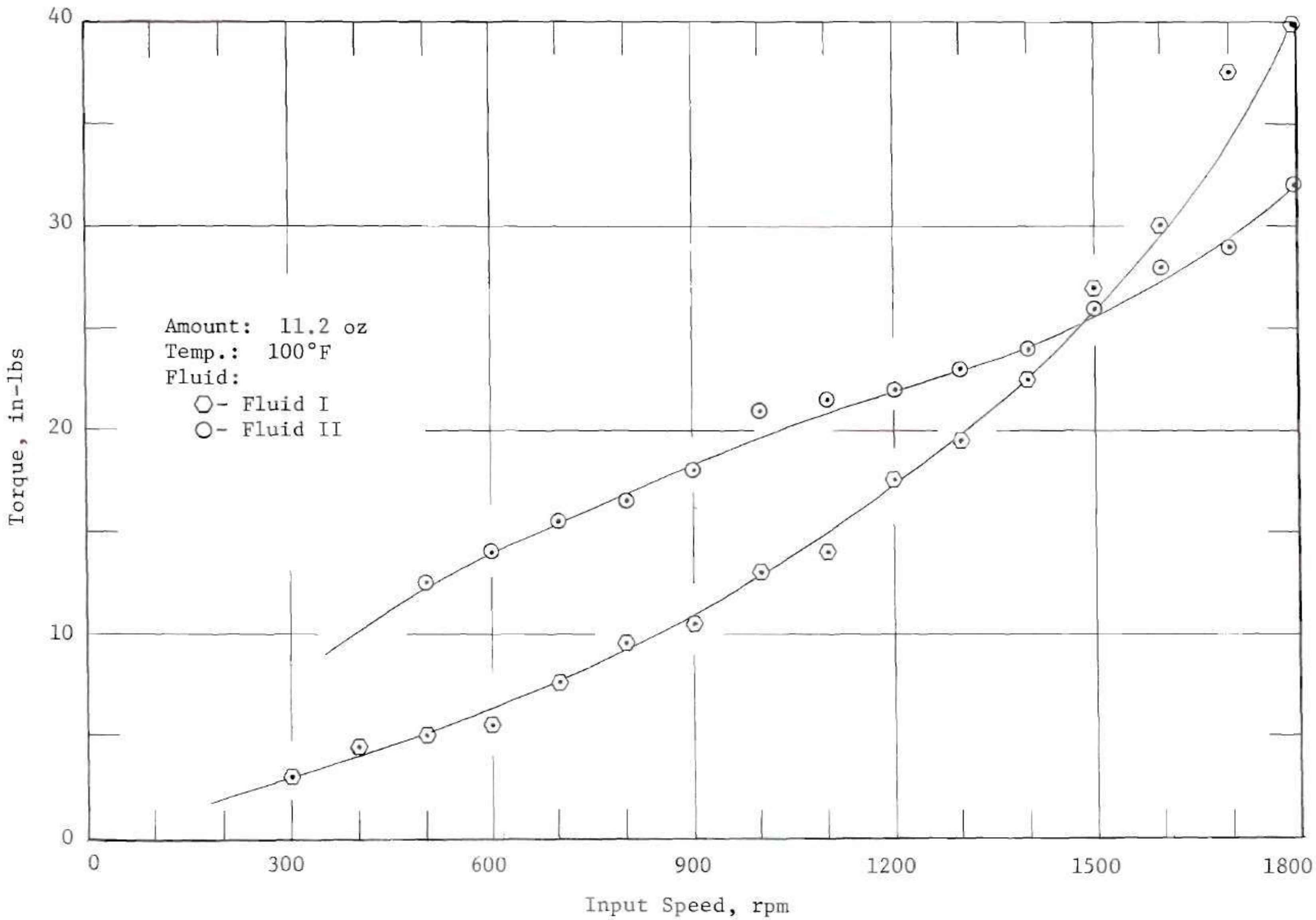


Figure 34. Comparative Drag Torque for Fluids I and II.

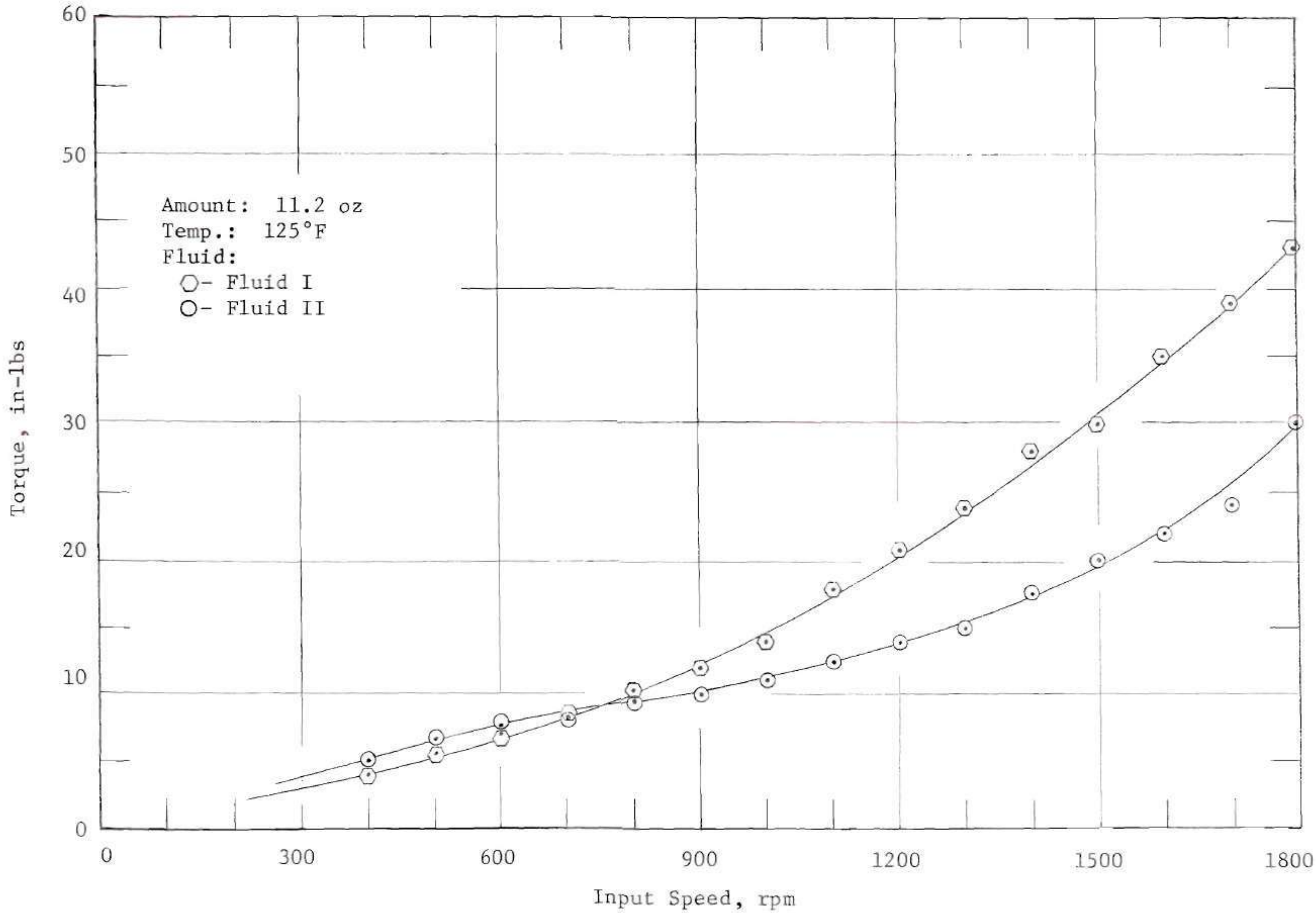


Figure 35. Comparative Drag Torque for Fluids I and II.

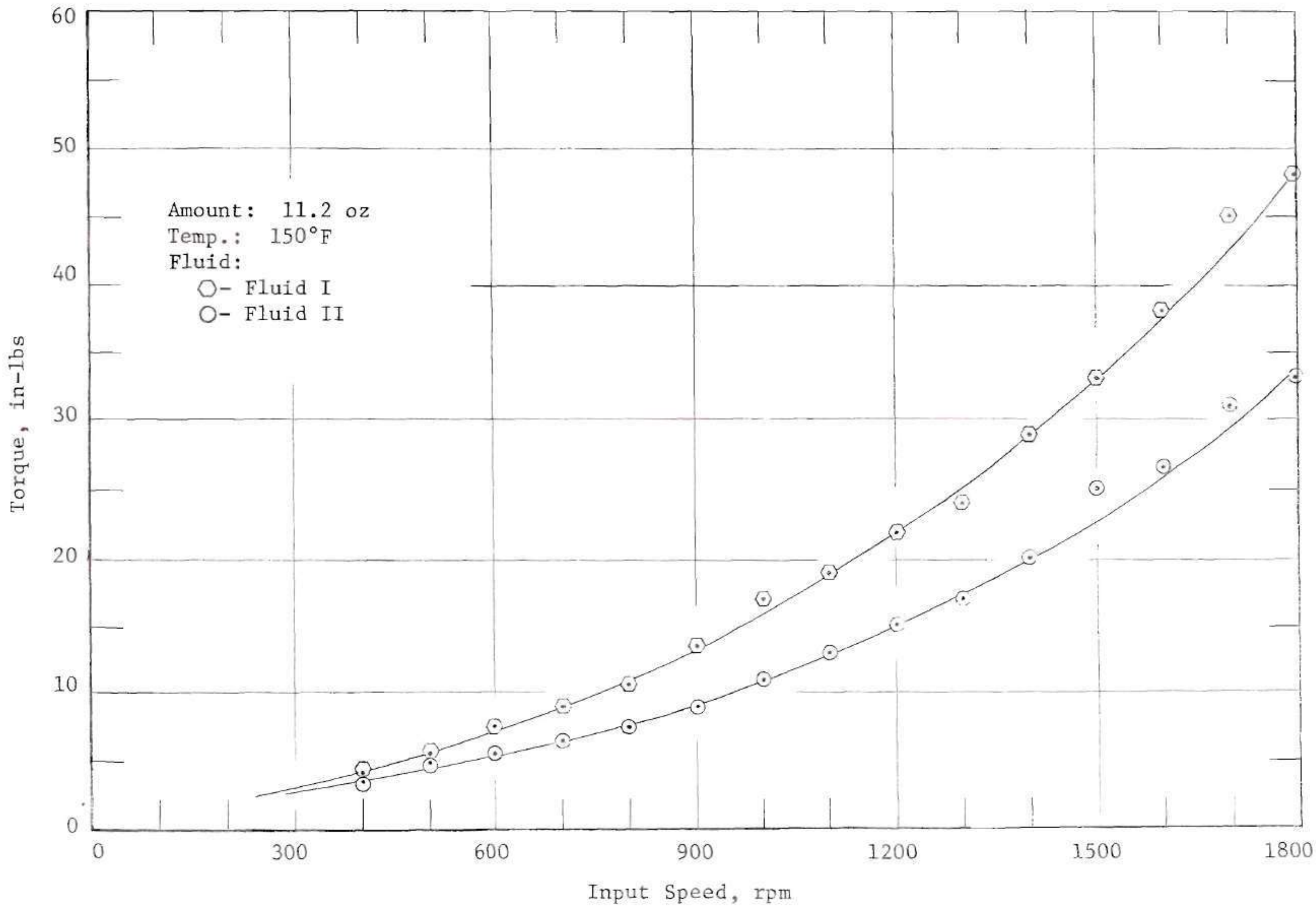


Figure 36. Comparative Drag Torque for Fluids I and II.

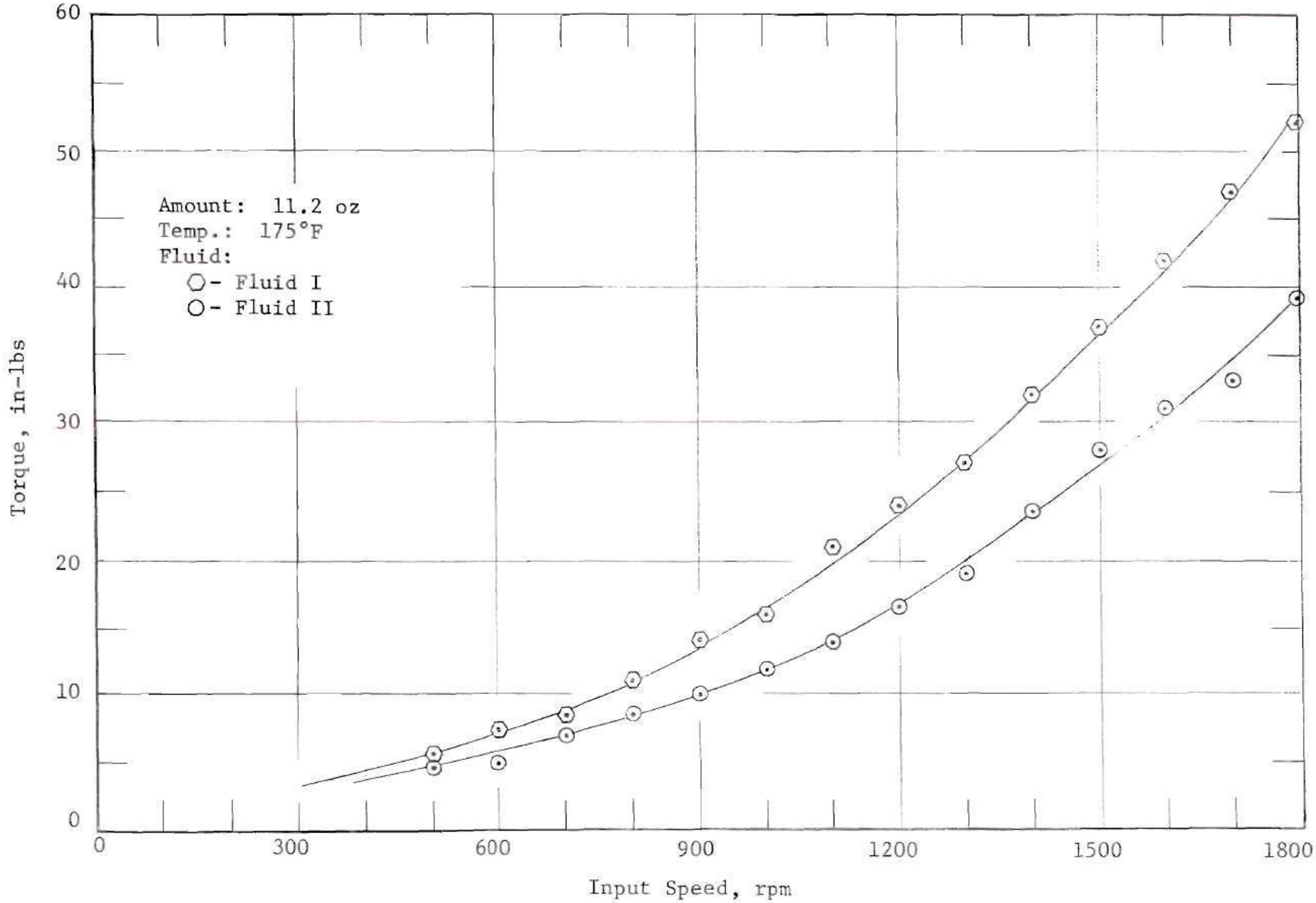


Figure 37. Comparative Drag Torque for Fluids I and II.

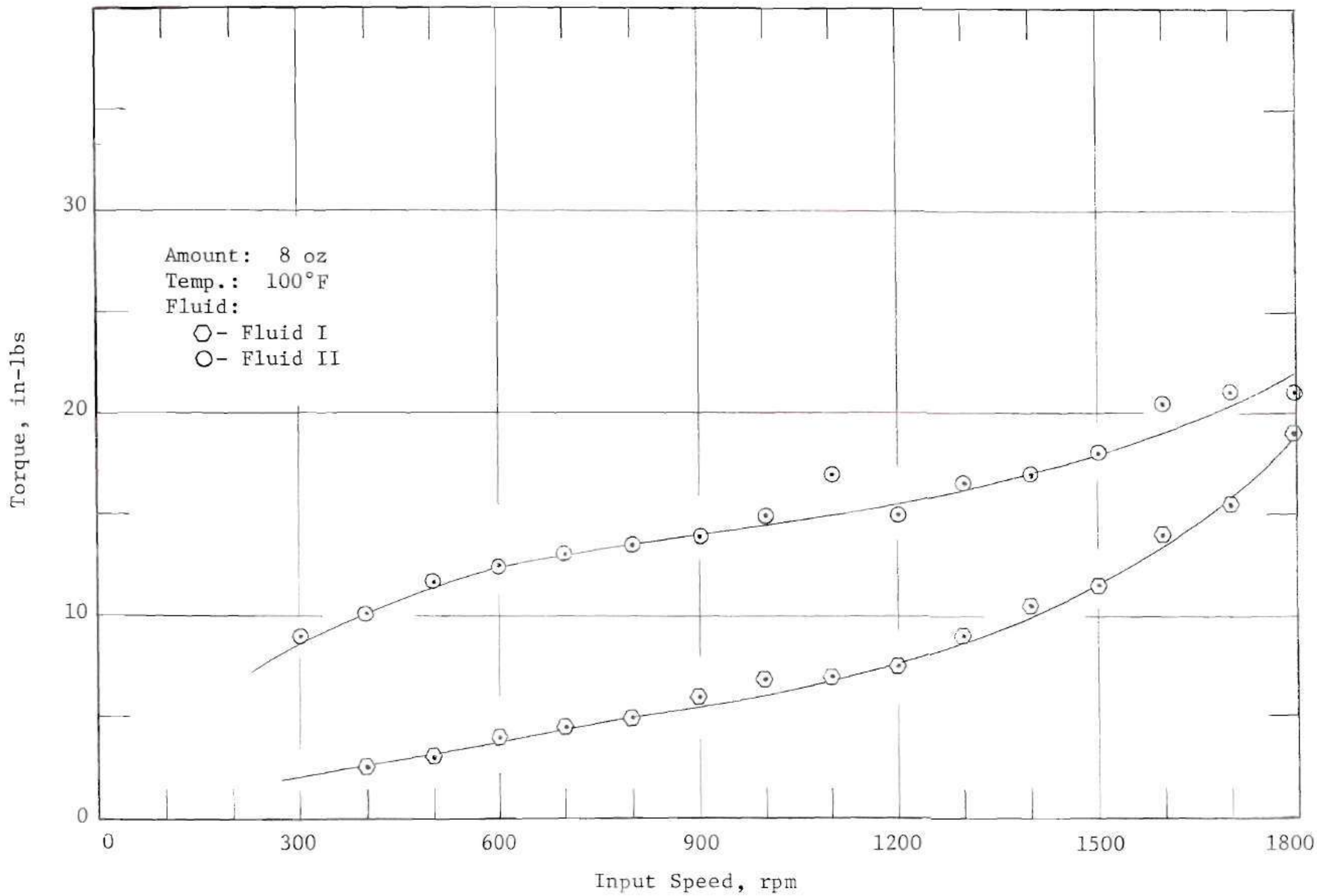


Figure 38. Comparative Drag Torque for Fluids I and II.

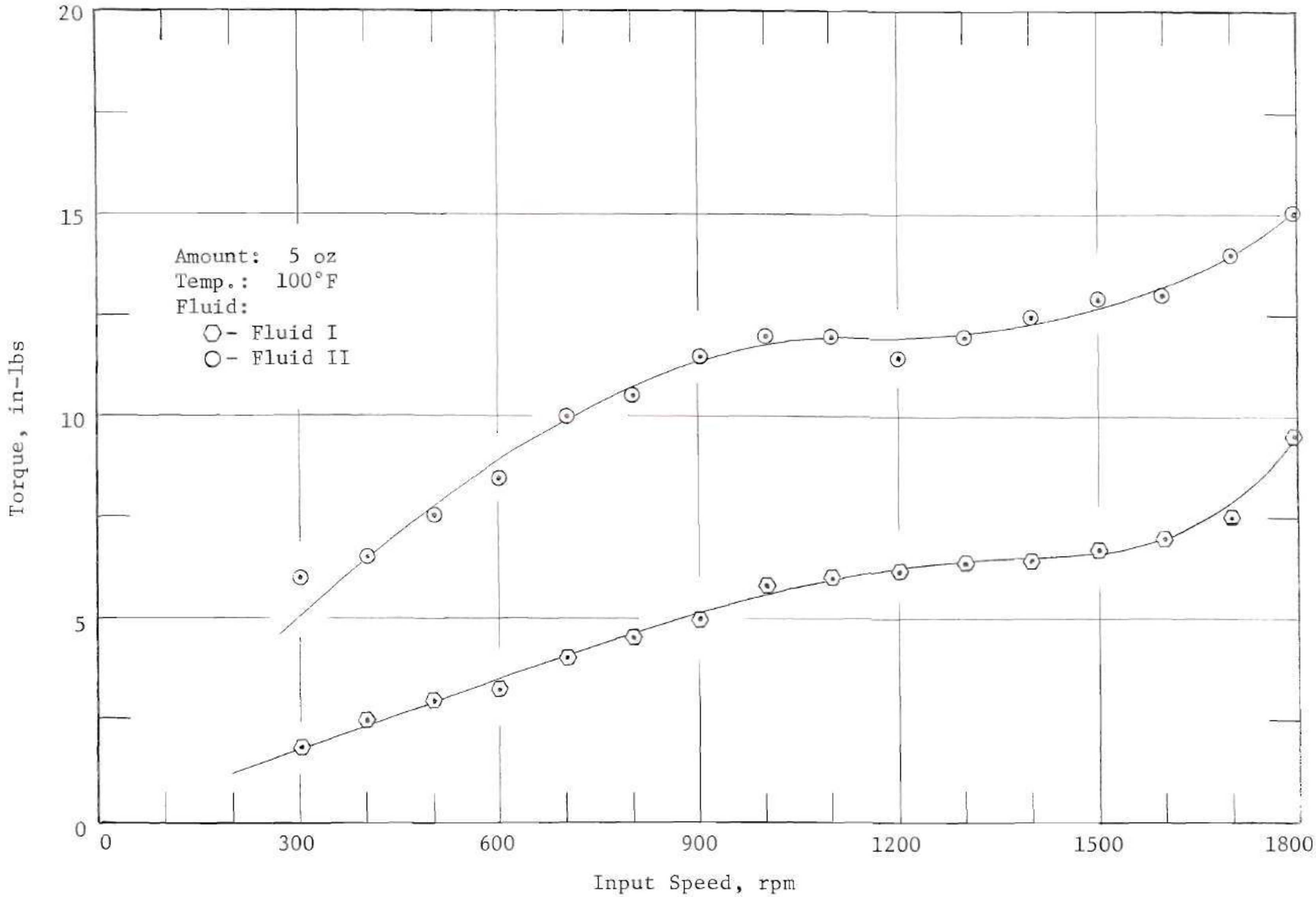


Figure 39. Comparative Drag Torque for Fluids I and II.

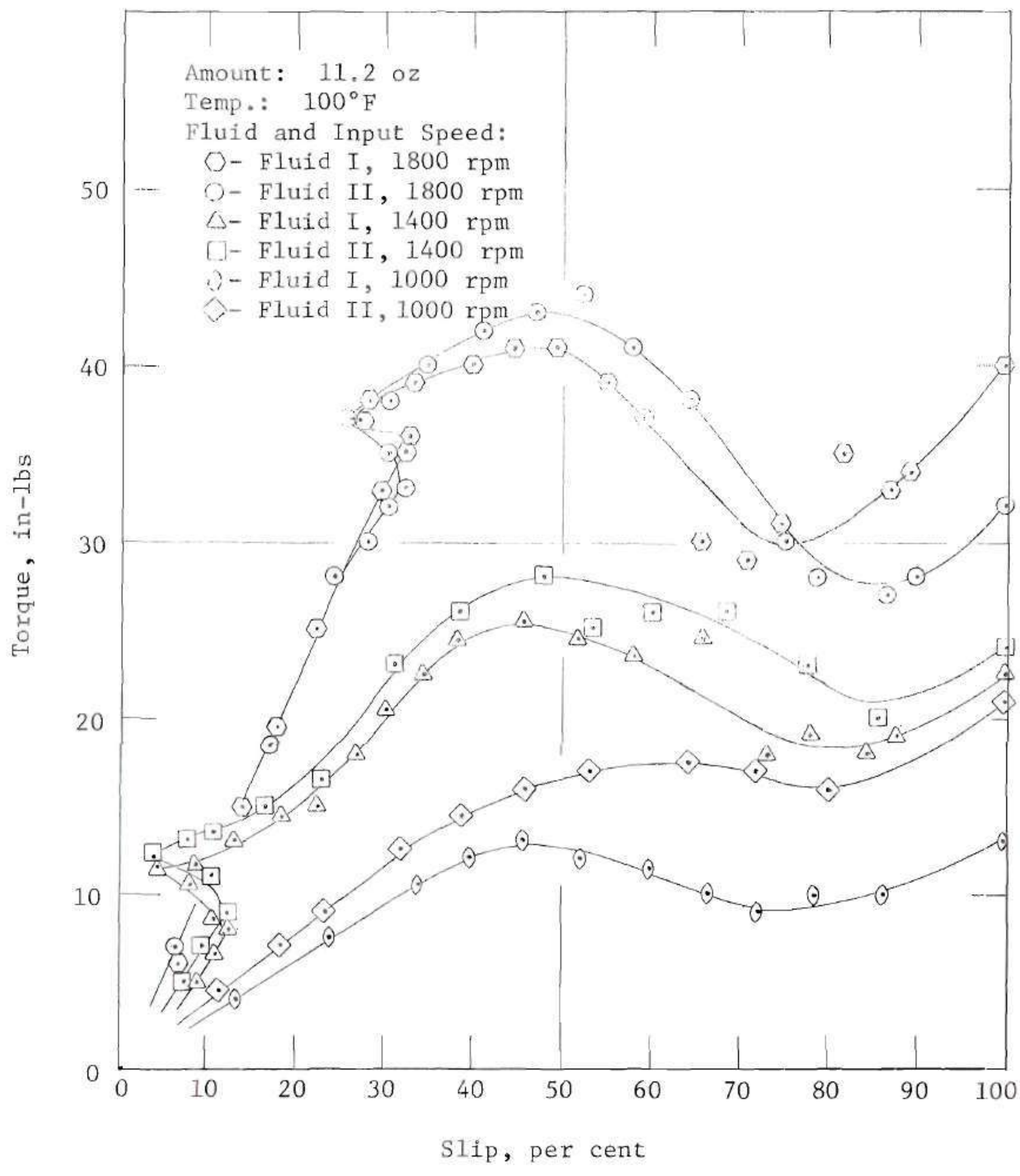


Figure 40. Torque-Slip Curves for Fluids I and II.

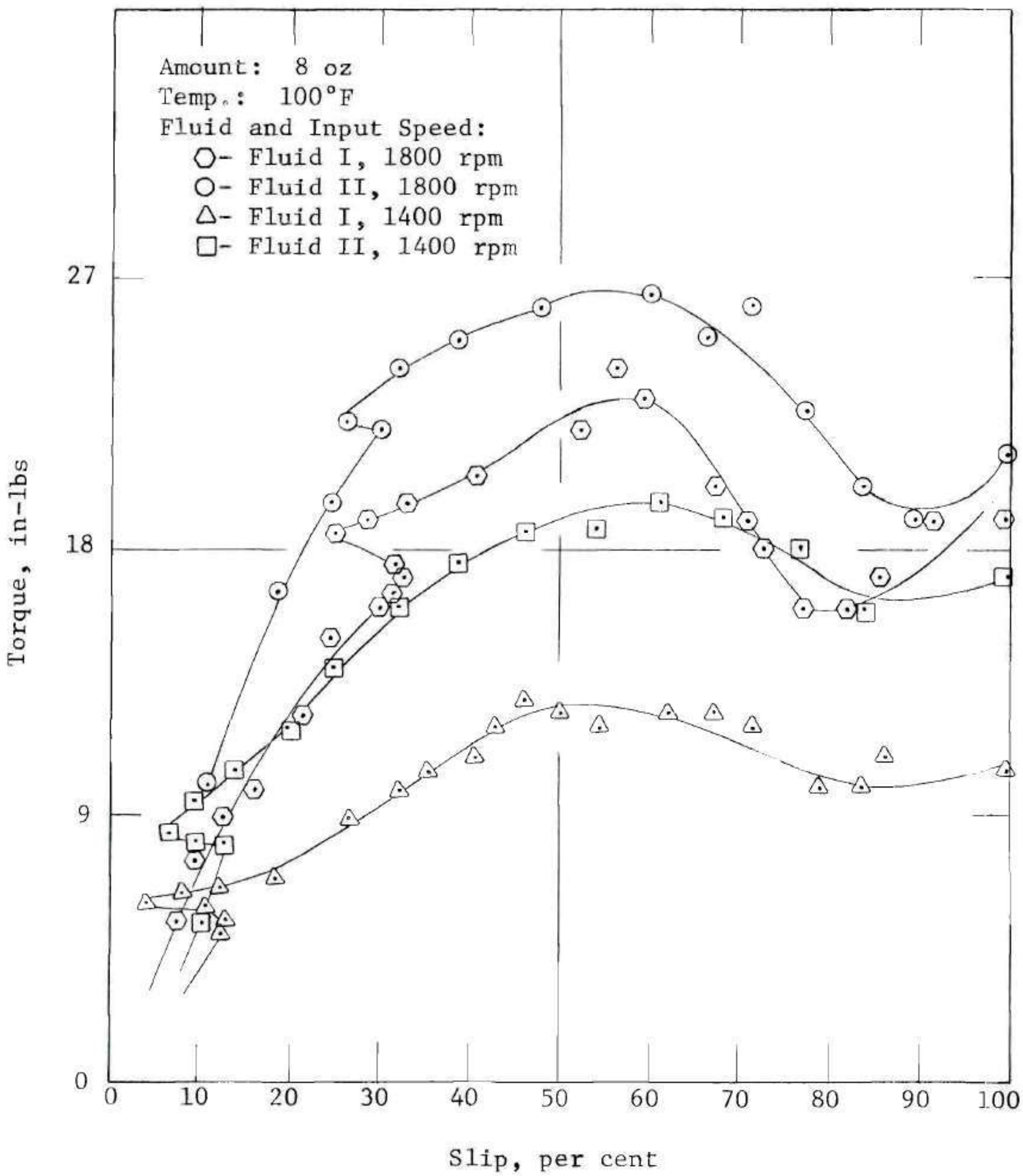


Figure 41. Torque-Slip Curves for Fluids I and II.

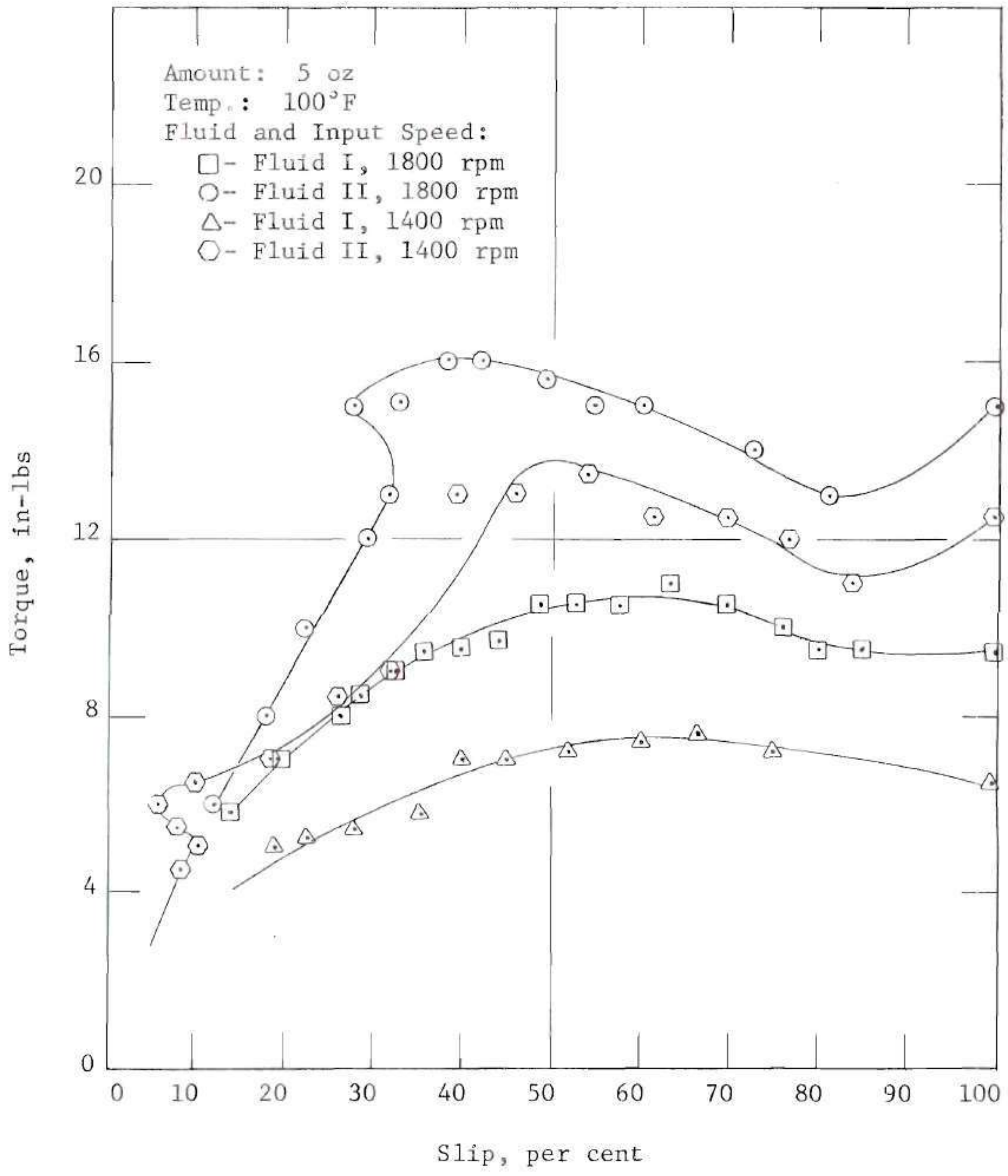


Figure 42. Torque-Slip Curves for Fluids I and II.

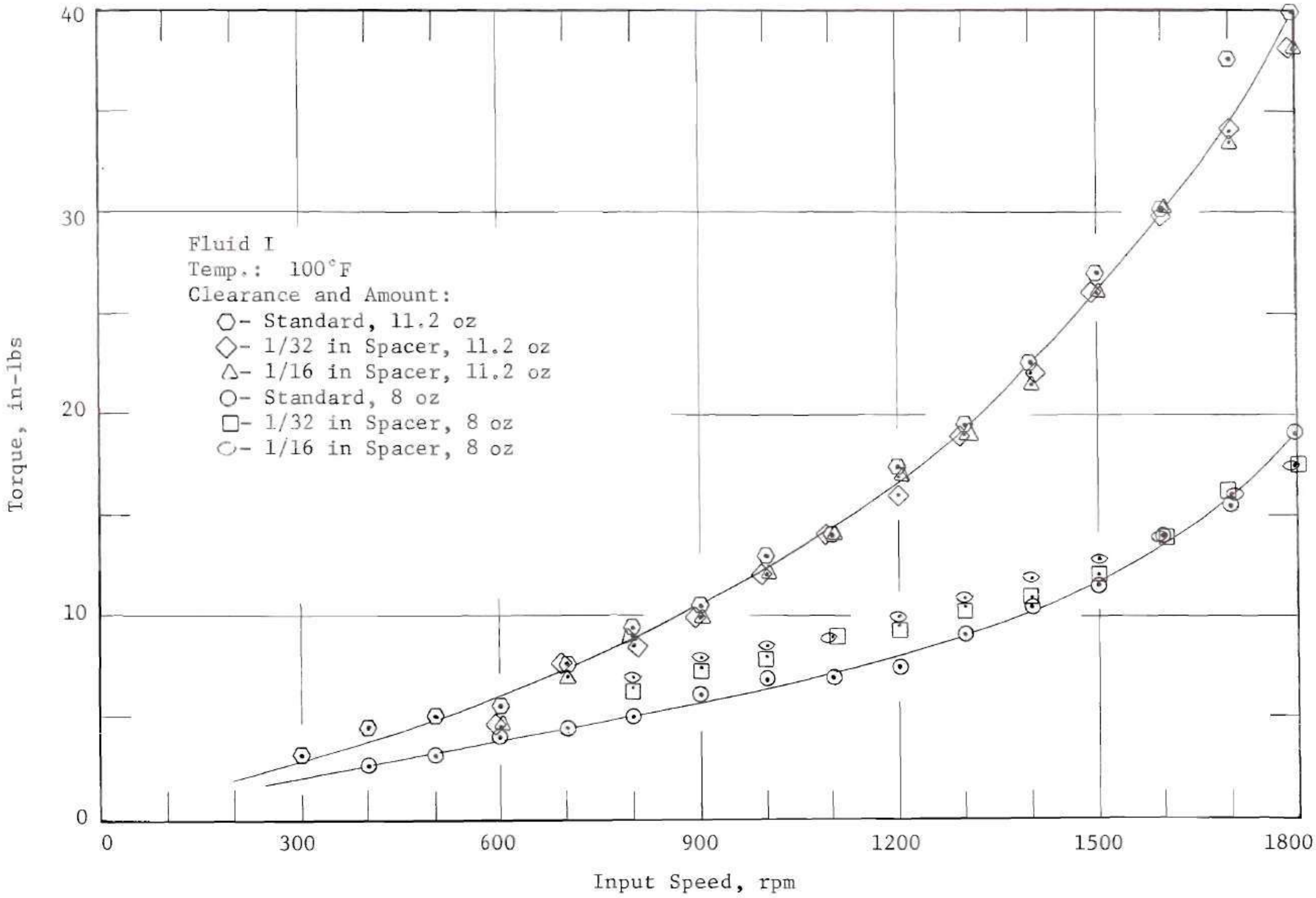


Figure 43. Drag Torque Showing Effects of Clearance Changes.

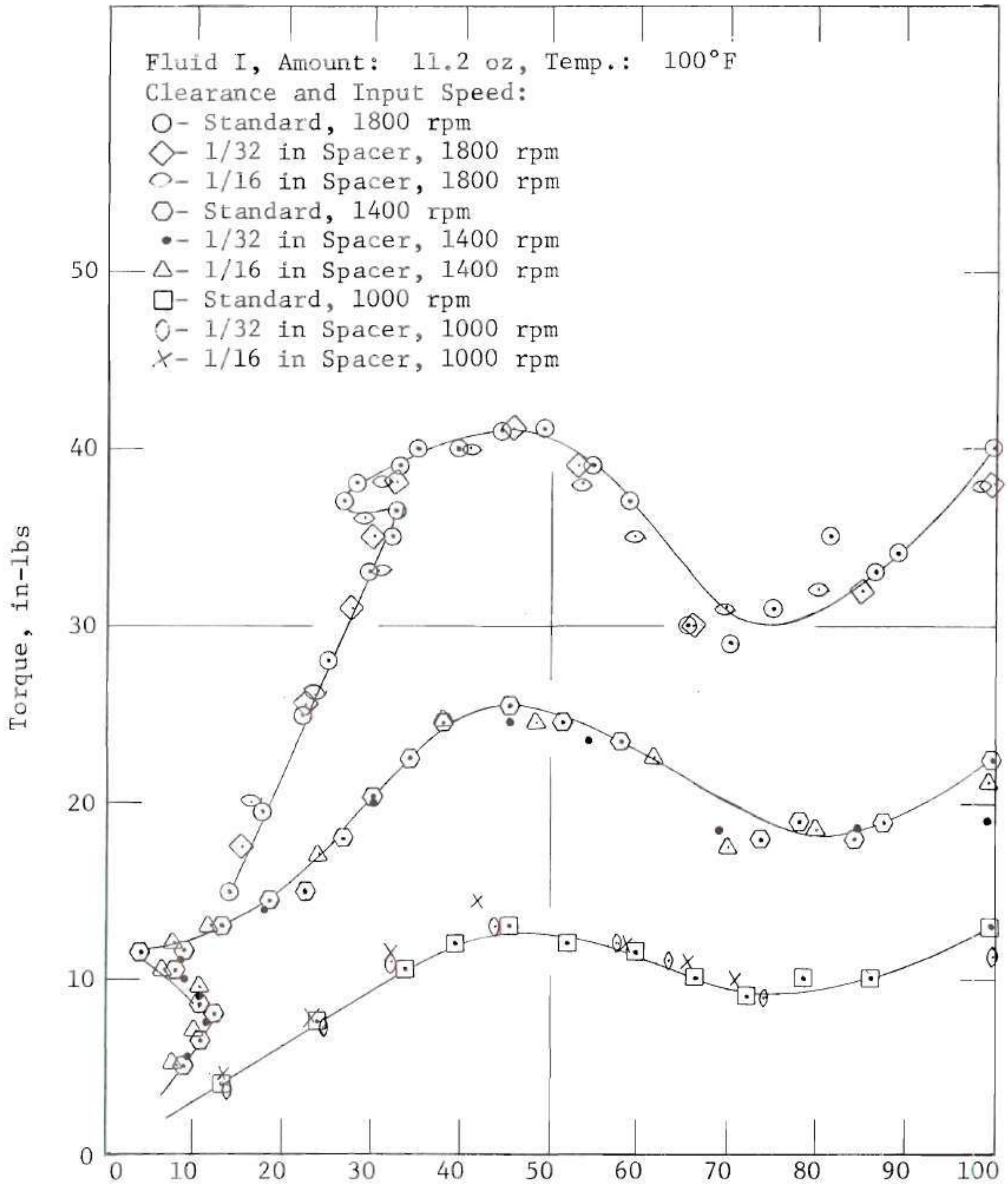


Figure 44. Torque-Slip Curves Showing Effects of Clearance Changes.

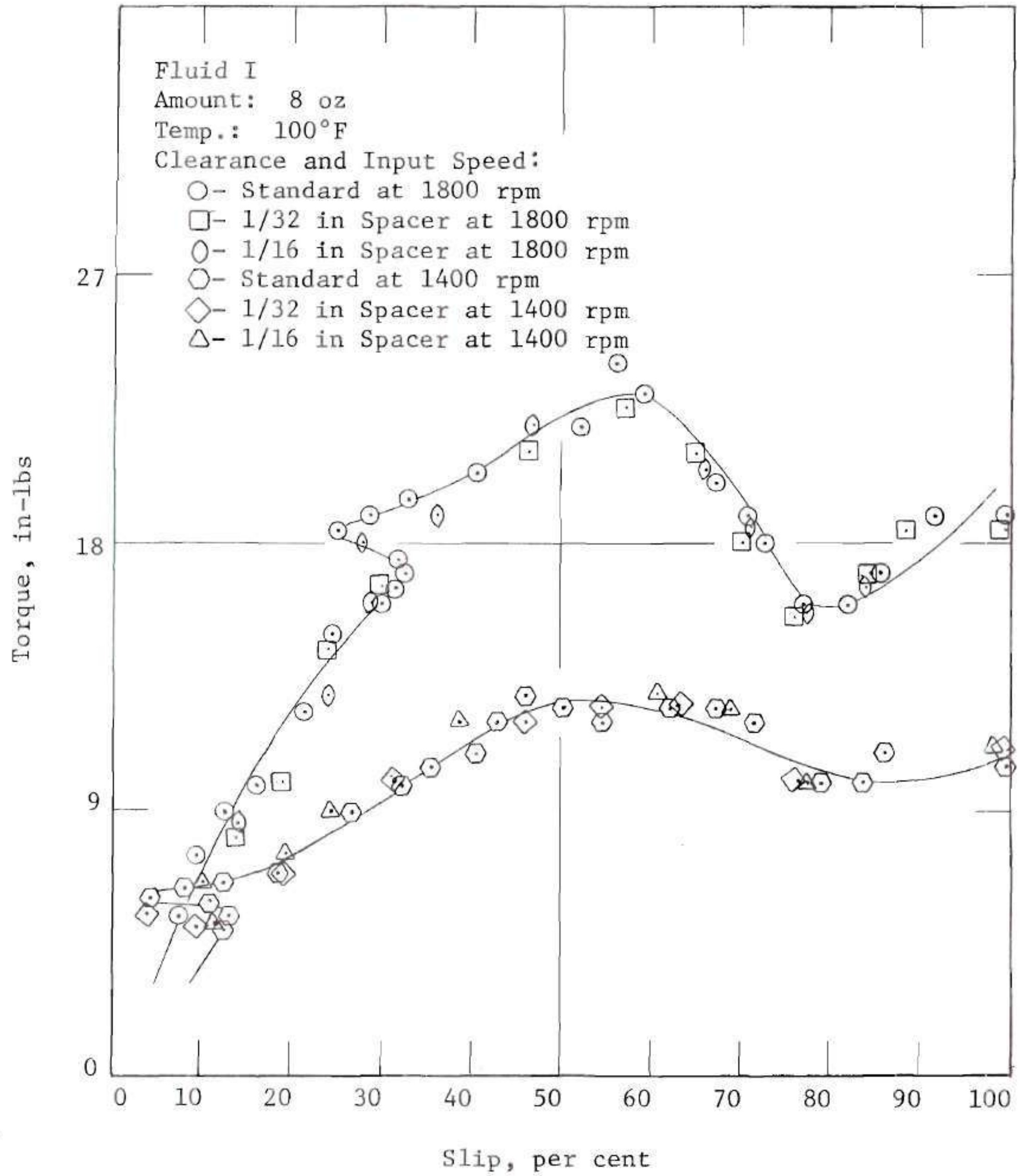


Figure 45. Torque-Slip Curves Showing Effects of Clearance Changes.

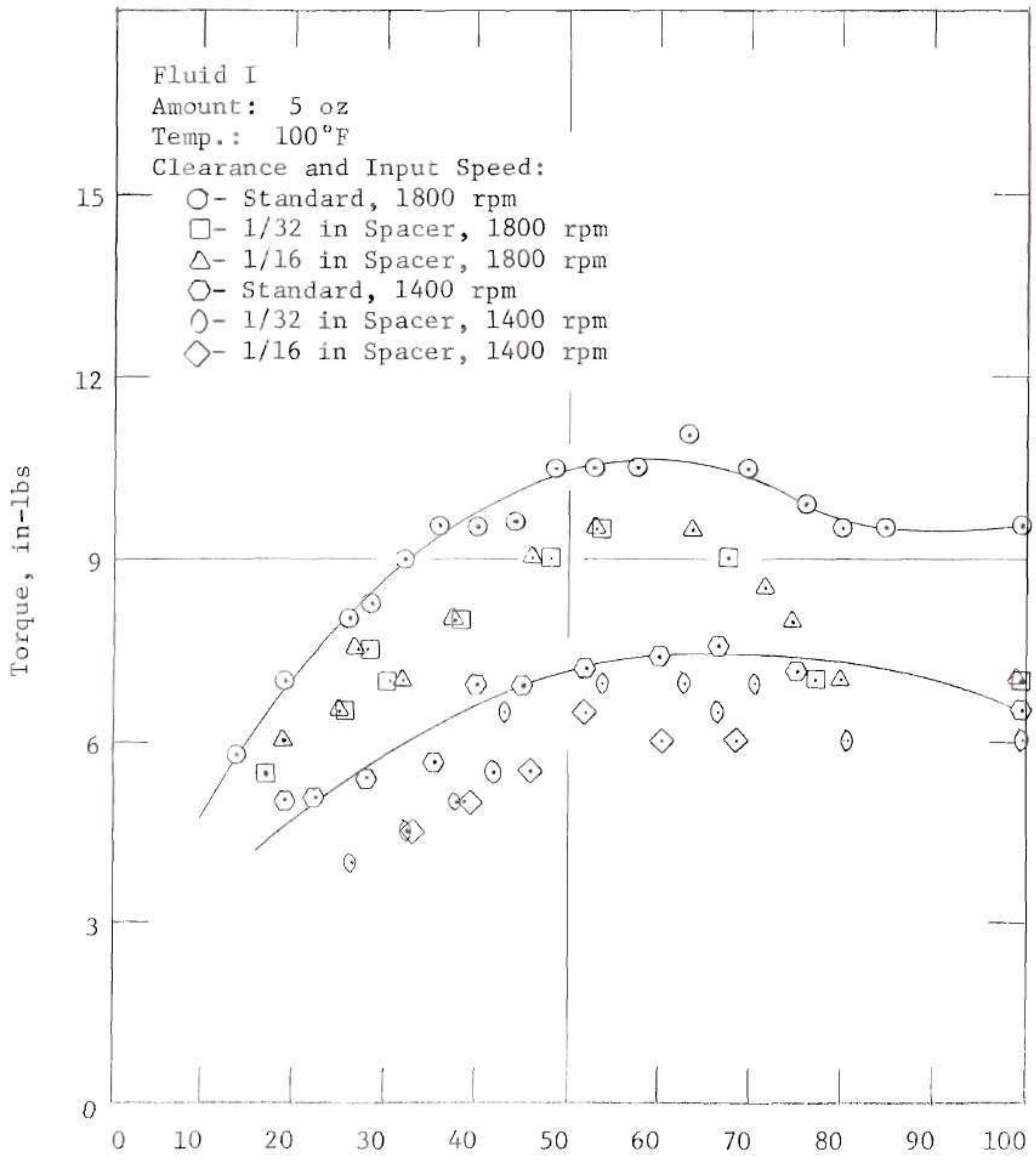


Figure 46. Torque-Slip Curves Showing Effects of Clearance Changes.

APPENDIX C

THEORETICAL CONSIDERATIONS OF FLUID CLUTCH

Efficiency

In Chapter II an equation for theoretical torque was derived. It showed that the mutual reaction torque between the impeller and turbine was equal and opposite.

Since torque times angular velocity is power, the overall efficiency, η, equals

$$\eta = \frac{T_T \ \omega_T}{T_I \ \omega_I}$$

where the subscript "I" refers to the impeller and "T" refers to the turbine.

Now, $\omega_T = (100\text{-}S)\omega_I$, where S is the per cent slip, and primary torque = secondary torque, so that

$$\eta = \frac{T(100-S)\omega_I}{T \ \omega_I} = 100-S$$

Therefore, the overall efficiency varies linearly with the slip.

Friction and Shock Losses

Power losses in the fluid clutch consist of bearing and windage losses, friction losses, and entrance shock losses. Bearing and windage losses constitute only a small fraction of the total losses and will be neglected.

Friction Losses

Friction losses are dependent on the curvature of the circuit, the roughness of the walls and vanes, the number and thickness of the vanes, the viscosity and density of the fluid, and whether the flow is viscous or turbulent. The complexity of the fluid flow limits the accuracy of any simple equation so the following is only an approximation.

In general practice, the frictional head loss is based on the Euler equation, i.e., taken as being proportional to the square of the circulation velocity of the fluid. The head loss expression for frictional resistance is then

$$h_f = \xi \frac{V_c^2}{2g}$$

ξ is a dimensionless coefficient of fluid friction.

Entrance Shock Losses

Because of the difference in the speeds between the impeller and turbine, discontinuities occur in the fluid velocities as they pass from one member to the other, and eddying and turbulent losses result. Also, the fluid flow between the vanes is not parallel motion and the fluid changes direction between two vanes in the impeller or turbine. The resulting losses are known as entrance shock losses. This loss is generally taken as proportional to the square of the differences in the abrupt changes in the tangential velocities at the turbine and impeller entrance. Referring back to Figure 3, the head loss equation is as follows:

$$h_s = \frac{(\Delta V u_o - \Delta V u_i)^2}{2g}$$

where

$$\Delta V_{u_o} = V_{u_2} - V_{u_3}$$

$$\Delta V_{u_i} = V_{u_1} - V_{u_4}$$

Actual shock losses are smaller than the theoretical loss. Therefore, the theoretical value can usually be multiplied by a shock factor which ranges from around 0.5 to 0.7 (34).

Axial Forces

For large diameters and high speeds, the forces tending to separate the two members and the impeller and casing can be significant. Assume the clutch in Figure 47 is in operation. The resulting axial forces arise from two effects: the pressure created by the centrifugal force on the fluid and the pressure caused by the fluid circulation.

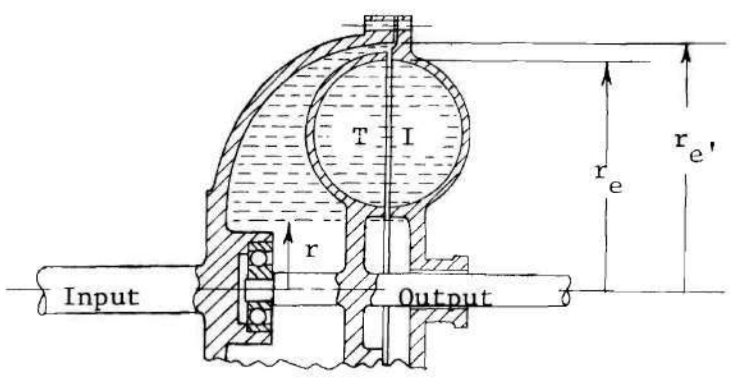


Figure 47. Schematic Showing Radii Used in Axial Force Equations.

Now, the axial force separating the impeller and housing is

$$F_I = F_{S_I} + F_d$$

where

F_{S_I} = Force created because of the hydrostatic pressure caused by centrifugal effects

F_d = Force due to hydrodynamic pressure caused by the fluid circulation

Since the impeller and casing rotate at a higher speed than the turbine, the hydrostatic pressure on the back of the turbine is greater than on the inside. Hence, the force on the turbine is as follows:

$$F_T = (F_{st_e} - F_{st_i}) - F_d$$

where

$(F_{st_e} - F_{st_i})$ = net axial force because of the external and internal hydrostatic pressure

F_d = force resulting from hydrodynamic pressure created by fluid circulation

By using the pressure distribution of a fluid rotating with its container to determine the hydrostatic forces and the momentum relation for steady flow to determine hydrodynamic forces, the axial forces can be derived. These forces as derived by Wolf (35) are

$$F_I = \frac{\rho}{g}\frac{\Pi}{4}\omega_I^2\{r_e^4[1 - (\frac{1+e}{2})^2] + r_{\acute{e}}(\frac{1+e}{2})^2 - r^4\} + \frac{2WV_c}{g}$$

$$F_T = \frac{\rho}{g}\frac{\Pi}{4}(r_e^4 - r^4)\ \omega_I^2[(\frac{1+e}{2})^2 - e^2] - \frac{2WV_c}{g}$$

where

W = Flow rate

V_c = Circulation velocity

ρ = Density

ω_I = Impeller angular velocity

e = Ratio of turbine speed to impeller speed

The radii r_e, $r_{\acute{e}}$, and r are as shown in Figure 47.

Heat Transfer

The cycle rate of a fluid clutch depends on the time response and the heat generated during and after engagement. Time response is a physical characteristic of the clutch, whereas, the heat dissipated within the clutch is a function of the slip. Since the efficiency is 100 minus the per cent slip, the power lost to heat is the input power times slip. Let P_I = input horsepower, then the heat flow, Q, due to slip is

$$Q = 42.4\ P_I\ S \quad \frac{\text{Btu}}{\text{min}}$$

In space this heat must be dissipated from the clutch by radiation or conduction since no atmosphere is normally present and the gravitation force is absent. Both of these factors are necessary for convective heat transfer.

Since the traction clutch is supported only by two shafts, radiation is the major mode of heat transmission. Depken (36) presented a detailed discussion of a clutch without connections contained within a surrounding container whose temperature was constant. He assumed uniform irradiation for both clutch and surroundings and a gray body. The heat transfer per unit area of the clutch by radiation was given as follows:

$$\frac{Q}{A_1} = \frac{\sigma(\overline{T}_1^{\,4} - \overline{T}_2^{\,4})}{\frac{1}{\varepsilon_1} + \frac{A_1}{A_2}\left(\frac{1}{\varepsilon_2} - 1\right)}$$

where

A = Area

ε = Emissivity

$\overline{T}$ = Temperature

σ = Stefan-Boltzmann constant

Subscript "1" and "2" refer to the clutch and surrounding, respectfully. Emissivity, ε_1, was assumed equal to ε_2, and curves were presented for heat transfer per unit area for emissivity values from 0.1 to 1.0 and for area ratios from 0.0 to 1.0.

The adjustable speed clutch offers a different situation because of its reservoir, which generally serves as a base for the clutch, and the cooler. In this case both radiation and conduction apply. By positioning the cooler away from the clutch, it may be possible to use the fuel on board as a heat sink (37) or use thermally controlled baffles to change the effective emissivity of a radiator (38).

Conductive heat transfer between the reservoir and its support should also be considered. Anderson (39) presented a detailed discussion of heat transfer by conduction in a vacuum, taking into account the thermal contact conductance (ratio of heat transferred across a joint per unit time, per unit area, per unit temperature drop) between the mating surfaces. A short summary follows.

Due to irregularities in surface finishes of metals, the actual physical contact area between any two surfaces is usually a small fraction of the total area. Heat transfer (for vacuum conditions) across the joint occurs then by thermal radiation and conduction through the actual contact points. Therefore, thermal contact conductance depends on the actual contact area and the properties of the metals. It reduces the heat transfer. Various methods to improve this factor are in use, such as: 1) increasing the contact pressure between the surfaces, 2) reducing surface roughness, 3) improving flatness of mating parts, and 4) using soft shim materials between surfaces. A combination of rubber and oil was reported to have greatly increased the contact conductance.

Anderson presented an example using this factor which reduced the heat transfer across a joint by approximately 94 per cent from the value obtained assuming perfect contact.

BIBLIOGRAPHY

Literature Cited

1. Wing Commr. T. R. Cave-Browns-Cave, Discussion on "Recent Developments in Hydraulic Couplings", Proceedings of the Institution of Mechanical Engineers, Vol. 130, April 26, 1935, p. 170.

2. F. W. Moss, Ibid., p. 170.

3. Dr.-Ing. Herrmann Föttinger, Ibid., pp. 158-61.

4. Dr.-Ing. Herrmann Föttinger, Discussion on "Some Problems in the Transmission of Power by Fluid Couplings", Proceedings of the Institution of Mechanical Engineers, Vol. 139, April, 1938, pp. 158-61.

5. Harold Sinclair, "Recent Developments in Hydraulic Couplings", Proceedings of the Institution of Mechanical Engineers, Vol. 130, April 26, 1935, pp. 80-7.

6. Ibid., pp. 75-158.

7. Harold Sinclair, "Some Problems in the Transmission of Power by Fluid Couplings", Proceedings of the Institution of Mechanical Engineers, Vol. 139, April, 1938, pp. 83-157.

8. "Chrysler is First American Car to Adopt the 'Fluid Drive'", Automotive Industries, Vol. 79, No. 26, 1938, pp. 819-20.

9. Austin M. Wolf, "Fluid Flywheel Marks New Era in Auto Transmission", Machine Design, Vol. 11, No. 1, 1939, pp. 23-4, 38.

10. A. H. Allen, "Beneath that Streamlined Exterior", Machine Design, Vol. 12, No. 11, 1940, p. 48.

11. "New Borg & Beck Fluid Coupling", Automotive Industries, Vol. 85, No. 1, 1941, pp. 34, 66.

12. Scanning the Field for Ideas, "Flexible Blades Limit Torque", Machine Design, Vol. 33, No. 26, 1961, p. 101.

13. Twin Disc Clutch Co., "Means of By-passing Coupling at Low Speeds for Supercharger", United States Patent 2,326,570.

14. J. Bilton, "Emptying Valves for Fluid Couplings", The Engineer, Vol. 217, No. 5644, 1964, pp. 576-7.

15. "Noteworthy Patents", Machine Design, Vol. 20, No. 7, 1948, p. 172.

16. The Texas Company, "Fluid Couplings and Torque Converters for Industrial Equipment", Lubrication, Vol. 36, No. 2, 1950, pp. 13-18.

17. Power in Textiles, "Smooth Starting with Electric Motors", The Textile Weekly, Vol. 62, June 1, 1962, pp. 1253-4.

18. Norman Peach, "Cushioned Starts for Tough Loads", Power, Vol. 104, No. 11, 1960, pp. 80-2.

19. R. D. O'Neil, Personel Correspondence, American-Standard Industrial Division.

20. Ing. Maurizio Wolf, Stromungskupplungen und Stromungswandler - Berechnung und Konstruktion, Springer - Verlag, Berlin, Germany, 1962.

21. E. Kickbusch, Fottinger-Kupplungen und Fottinger-Getriebe, Berechnung und Konstruktion, Springer - Verlag, Berlin, Germany, 1963

22. P. M. Heldt, "A Mathematical Consideration of the Fluid Coupling", Automotive Industries, Vol. 82, No. 8, 1940, pp. 360-2, 384.

23. Robert E. Bruckner, "A Simple Method for Calculating Hydraulic Coupling Performance", Product Engineering, Vol. 13, No. 12, 1942.

24. V. L. Streeter, Fluid Mechanics, Second Edition, McGraw-Hill Book Co., Inc., New York, 1958.

25. H. A. Huebotter, "Discussion of a Mathematical Consideration of the Fluid Coupling", Automotive Industries, Vol. 82, No. 10, 1940, pp. 470-1.

26. W. F. Shurts, C. L. Sadler, "Fluid Power Transmissions", Product Engineering, Vol. 21, No. 3, 1950, pp. 132-3.

27. Ben Ragland, "Basic Facts About Fluid Couplings", Machine Design, Vol. 28, No. 9, 1956, pp. 108-16.

28. Robert W. Vance, "Introduction to Cryogenics", Machine Design, Vol. 36, No. 24, 1964, pp. 178-9.

29. R. G. Broadwell, R. A. Wood, "Titanium Alloys for Cryogenic Service", Materials Research & Standards, Vol. 4, No. 10, 1964, pp. 549-54.

30. J. E. Campbell, "Aluminum Alloys for Cryogenic Service", Materials Research & Standards, Vol. 4, No. 10, 1964, pp. 540-8.

31. Heinz Hanau, "Space Age Spin-Off ... Cryogenic Bearings", Machine Design, Vol. 36, No. 13, 1964, pp. 123-8.

32. D. Probert, P. Thomas, "Vessels for Low-Temperature Liquids", Chemical and Process Engineering, Vol. 45, December, 1964, pp. 679-85.

33. R. B. Scott, Cryogenic Engineering, D. Van Nostrand Co., Inc., Princeton, New Jersey, 1959, pp. 281, 283.

34. Ing. Maurizio Wolf, Stromungskupplungen und Stroungswandler - Berechnung und Konstruktion, Springer-Verlag, Berlin, Germany, 1962, p. 38.

35. Ibid., p. 79.

36. C. A. Depken, "Magnetic-Flux Clutches for Space Applications", Master's Thesis, Georgia Institute of Technology, May, 1966.

37. G. L. Roth, O. L. Williamson, "How to Use Fuel as a Heat Sink", Space/Aeronautics, Vol. 33, No. 3, 1960, pp. 56-60.

38. R. A. Hanel, "Thermostatic Temperature Control of Satellites and Space Vehicles", American Rocket Society Journal, Vol. 29, 1959, p. 358.

39. R. I. Anderson, "Mechanical-Contact Clutches for Space Applications", Master's Thesis, Georgia Institute of Technology, February, 1966.

Other References

Alison, N. L., "Fluid Transmission of Power", SAE Journal, Vol. 48, No. 1, 1940, pp. 1-9.

Alison, N. L., Olson, R. G., and Nelden, R., "Fluid Couplings", Machine Design, Vol. 12, No. 8, 1940, pp. 45-7.

Alison, N. L., Olson, R. G., and Nelden, R. M., "Hydraulic Couplings for Internal-Combustion-Engine Applications", Trans. ASME, Vol. 63, Feb. 1941.

Alper, R. H., "Cryogenically Stretch-Formed Type 301 Stainless Steel for Cryogenic Service", Materials Research & Standards, Vol. 4, No. 10.

Annett, F. A., "Hydraulic Drives", Power, Vol. 89, No. 3, 1945, pp. 163-6.

Annett, F. A., "Industrial-Power Clutches", Power, Vol. 92, No. 2, 1948, p. 94.

Bachmann, R. W., "Clutch-Fluid Coupling-Hydraulic Torque Converter", Mechanical Engineering, Vol. 81, No. 12, 1959, pp. 67-9.

Barron, R. F., "Superinsulations", Machine Design, Vol. 33, No. 5, 1961, pp. 116-20.

Binder, R. C., Advanced Fluid Dynamics and Fluid Machinery, Prentice-Hall, Inc., Englewood Cliffs, N. J., 1951, pp. 213-26.

Broadwell, R. G., and Wood, R. A., "Crystal Structure and Cryogenic Properties", Materials Research & Standards, Vol. 4, No. 10, 1964, pp. 523-4.

Briefing the Record, "Hydraulic Couplings", Mechanical Engineering, Vol. 60, No. 8, 1938, pp. 631-3.

Brunken, R., "The White Hydro Torque Drive", SAE Trans., Vol. 54, 1946, pp. 587-95.

Carmichael, C., "Specifying Variable-Speed Drives", Machine Design, Vol. 16, No. 6, 1944, pp. 107-12.

Eksergian, R., "The Fluid Torque Converter and Coupling", Journal of the Franklin Institute, Vol. 235, No. 5, 1943, pp. 441-78.

"Exhibits at the Engineering and Marine Exhibition", Engineering, Vol. 144, No. 10, 1937, pp. 367-8.

"Fluid Couplings and Torque Converters", Mechanical World & Engineering Record, Vol. 143, April, 1963, pp. 143-5.

Gagne, A. F., Jr., "Clutches", Machine Design, Vol. 24, No. 8, 1952, pp. 123-58.

Gibson, W. B., "Hydrodynamic Drives", Machine Design, Vol. 32, No. 7, 1960, pp. 108-12.

Gibson, W. B., "Selecting the Right Fluid Coupling for a Prime Mover", Product Engineering, Vol. 21, No. 10, 1950, pp. 136-40.

Gorsky, R. J., "Buick Switches the Pitch of its Dynaflow Transmission Stator Vanes to Give Better Efficiency at All Speeds", SAE Journal, Vol. 63, Oct., 1955, pp. 28-9.

Habicht, R. C., "Which Variable Speed Fluid Coupling", Product Engineering, Vol. 22, No. 10, 1951, pp. 125-8.

Heldt, P. M., "An Explanation of the Mode of Operation of Fluid Flywheels", Automotive Industries, Vol. 80, No. 4, 1939, pp. 109-111.

Heldt, P. M., Torque Converters or Transmissions, Third Edition, P. M. Heldt, Nyack, New York, 1947, pp. 76-88.

Ingenious Mechanical Movements, "Couplings of the Fluid or Hydraulic Type", Machinery, Vol. 47, No. 6, 1941, pp. 129-31.

Jania, Z. and Kushigian, A., "How Rotation Affects Hydraulic Pressures", Machine Design, Vol. 27, No. 2, 1955, pp. 180-8.

Karassik, I. J. and Carter, R., Centrifugal Pumps, F. W. Dodge Corp., New York, 1960, pp. 310-2.

Kelley, O. K., "Operating Principles of Fluid Couplings for Automotive Engines", Product Engineering, Vol. 29, No. 37, 1958, pp. E15-7.

Kimberly, A. E., "Fluid Couplings for Passenger Cars", SAE Quarterly Trans., Vol. 1, No. 4, 1947, pp. 577-83.

Leslie, R. L., "Petroleum-Base Hydraulic Fluids", Machine Design, Vol. 33, No. 16, 1961, pp. 114-7.

Lezard, C. V., "Hydraulic Variable-Speed Drives", Engineers' Digest, Vol. 25, No. 12, 1964, pp. 99-164.

McCausland, R. D., "Hydraulic Couplings - I", Product Engineering, Vol. 12, No. 1, 1941, pp. 16-9.

McCausland, R. D., "Hydraulic Couplings - II", Product Engineering, Vol. 12, No. 2, 1941, pp. 79-80.

Metaxes, T., "Clutches for Hard-to-Start Loads", Mill & Factory, Vol. 69, No. 2, 1961, pp. 65-70.

Morrison, J. A., "Fluid Clutches for Electric Drives", Product Engineering, Vol. 25, No. 8, 1954, pp. 174-8.

"Non-Slip Fluid Coupling", Power & Works Engineering, Vol. 59, March, 1964, pp. 26-7.

Olson, R. G., "Hydraulic Couplings", Machine Design, Vol. 15, No. 4, 1943, pp. 137-8.

Ragland, B., "Adjustable-Speed Fluid Couplings", Diesel Power, Vol. 33, No. 11, 1955, pp. 50-1.

Rizzone, M. L., "Power Transmission on Modern Rotary Rigs", World Oil, Vol. 137, No. 7, 1953, pp. 124, 126, 128.

Rosebrough, J. D., "Fluid Coupling or Torque Converter?", Machine Design, Vol. 20, No. 4, 1948, pp. 149-52.

Seliber, J., "How Hydraulic Couplings Affect Gear and Shaft Selection", Product Engineering, Vol. 31, No. 34, 1960.

Sheperd, D. G., Principles of Turbomachinery, The Macmillan Co., New York, 1963, Chapter III.

Sinclair, H., "Fluid Couplings Give Automatic Torque Variation", Machine Design, Vol. 7, No. 7, 1935, pp. 29-32.

Sinclair, H., "Hydraulic Couplings Control Feed-Pump Speed", Power, Vol. 83, No. 9, 1939, pp. 66-7.

Spannhake, W., "Hydrodynamic Power Transmission for Motor Cars", SAE Trans., Vol. 45, No. 4, 1939, pp. 433-43.

Spannhake, W., "Hydrodynamic Torque Converter Supplements Fluid Coupling", Machine Design, Vol. 11, No. 5, 1939, pp. 39-41.

What's Happening in Research, "New Data on Bearings for Cryogenic Service", Product Engineering, Vol. 32, No. 35, 1961, p. 8.

Woodard, M., and Gibson, A., "Fluid and Dumping Couplings ... What Are They? ... and What Do They Do?", World Oil, Vol. 139, No. 1, 1954, pp. 158-9, 162, 166, 171.

Wislicenus, G. F., Fluid Mechanics of Turbomachinery, First Edition, McGraw-Hill Book Co., Inc., New York, 1947, pp. 372-96.